Vol. 12, No. 1 January-February, 1956

LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

In This Issue

Protective Oil Coatings (Practical Lubrication, No. 17)
by K. F. Schiermeier & R. W. Lewis 19

Wear Studies With Radioactive Gears
by V. N. Borsoff 24

Radioactive Study Of The Metal-Cutting Process, A
by I. Finnie & E. Rabinowicz 29

Rheodynamic Squeeze-Film, The
by F. Osterle, A. Charnes & E. Saibel 33

Study Of Fretting Wear In Mineral Oil, A
by D. Godfrey 37

Pumpability Of Aircraft Turbine Lubricants At Low Temperatures
by E. S. Starkman & J. H. Bridges 43

Progress Report On Silicone Lubricants, A
by H. F. Lamoreaux 48

Study Of The Sliding Of Metals, With Particular Reference To Atmosphere, A
by L. F. Coffin, Jr. 50

Advertisers' Index 72
ASLE 11th Annual Meeting Program (Tentative) 36a
ASLE Directory 65
Book Reviews, by A. R. Black, A. C. West & B. H. Jennings 28
Coming Events 42
Current Literature 62
Editorial, by J. W. Hopkinson, ASLE President 17
Engineering Societies Personnel Service, Inc. 69
Experiment & Experience ("Maintenance of Soluble Oil Emulsions in Large Systems," by O. L. Maag & H. T. Peeples) 9
Lube Lines, by A. F. Brewer 13
Lubrication Abstracts, compiled by Henry Brutcher 47
Lubrication in the News 62
Meet the Authors 5
Men of Lubrication ("Nikolay P. Petrov"), by F. R. Archibald 15
New Products 36
Patent Abstracts, compiled by Ann Burchick 66
Personals 42
Section News 23

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Table Of Contents

Protective Oil Coatings (Practical Lubrication, No. 17) by K. F. Schiermeier & R. W. Lewis 19

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Radioactive Study Of The Metal-Cutting Process, A by I. Finnie & E. Rabinowicz 29

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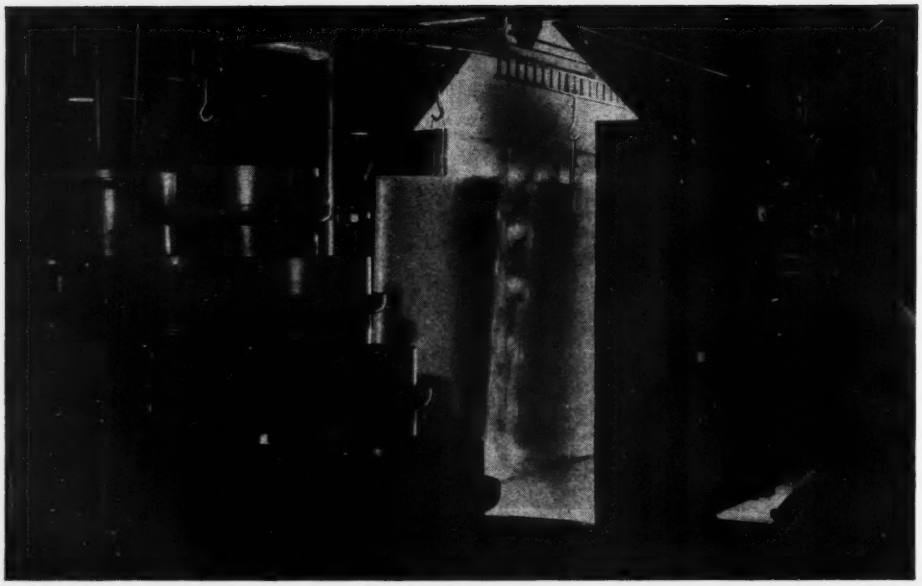

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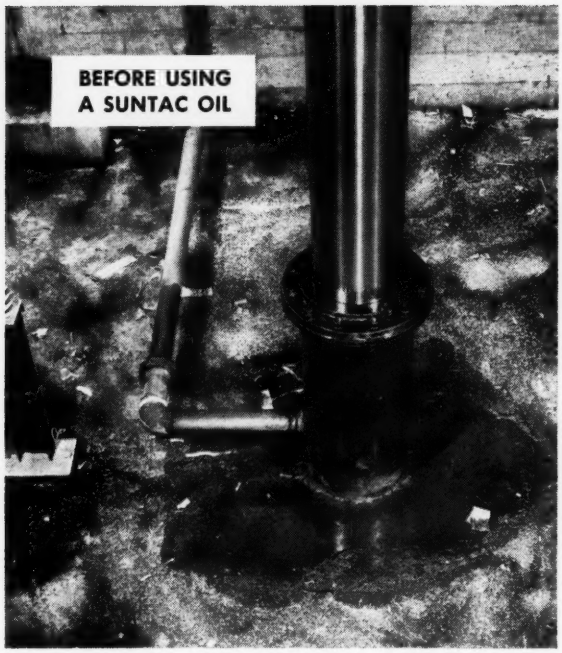

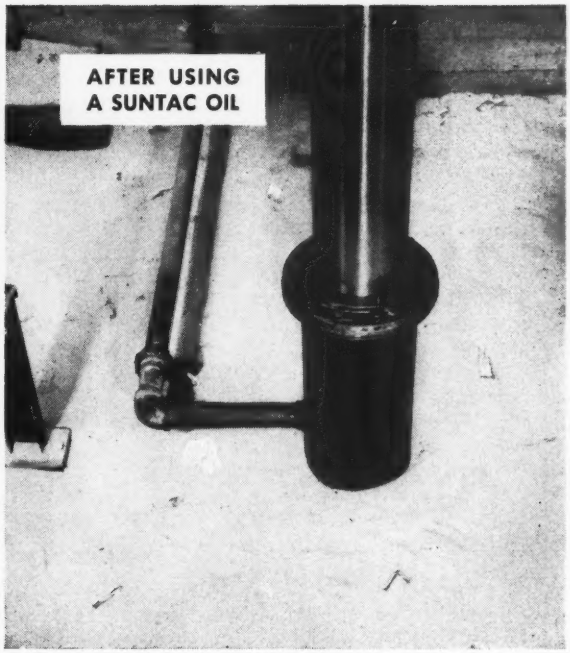

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Meet the Authors of the papers appearing in this issue of **Lubrication Engineering** (Vol. 12, No. 1). Mr. F. R. Archibald is the author of the biographies entitled "Men of Lubrication;" Messrs. I. Finnie & E. Rabinowicz; F. Osterle, A. Charnes & E. Saibel; K. F. Schiermeier & R. W. Lewis; and E. S. Starkman & J. H. Bridges, are co-authors of their respective papers, i.e. (1) "A Radioactive Study of the Metal-Cutting Process," (2) "The Rheodynamic Squeeze-Film," (3) "Protective Oil Coatings," and (4) "Pumpability of Aircraft Turbine Lubricants at Low Temperatures."

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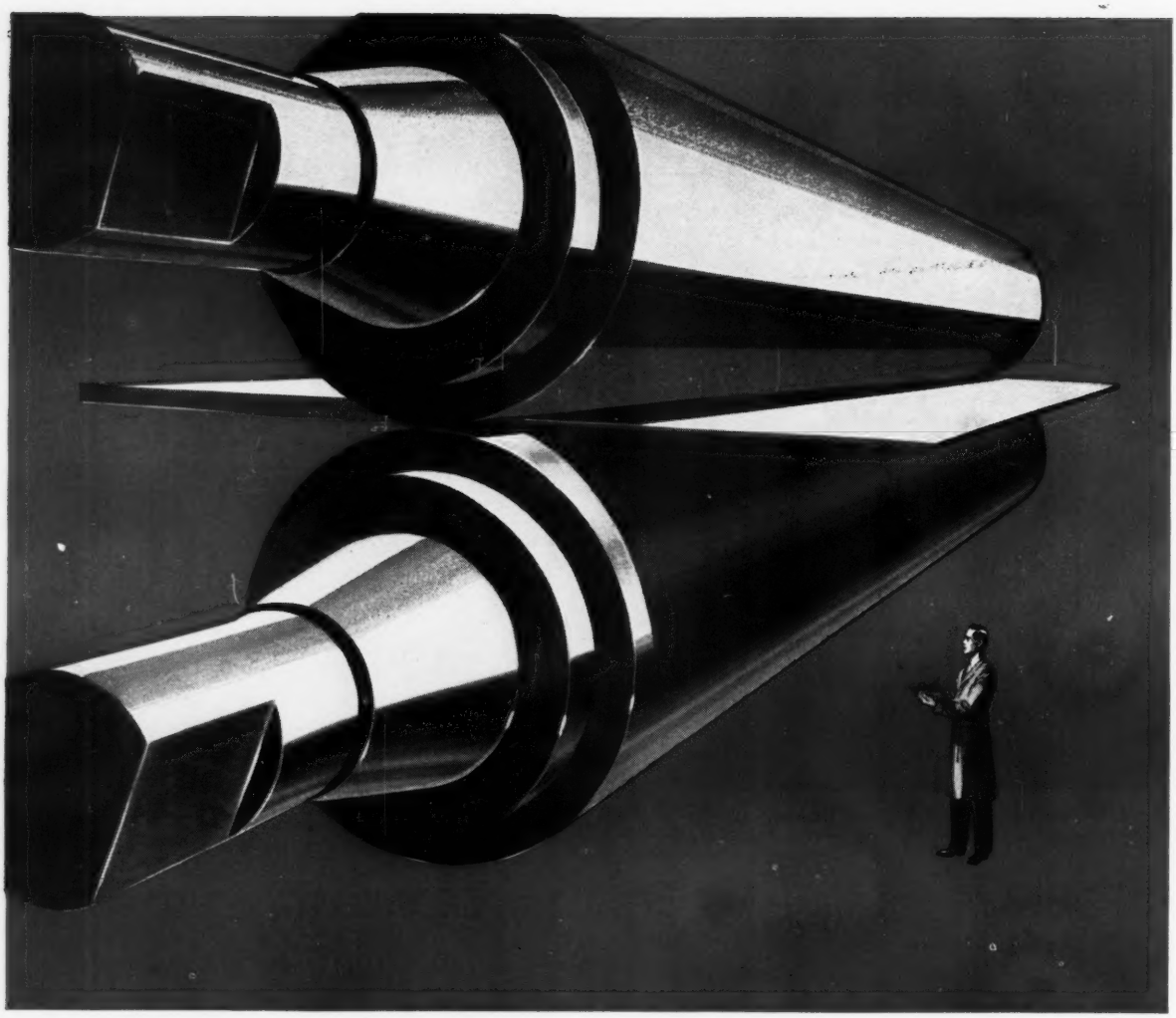

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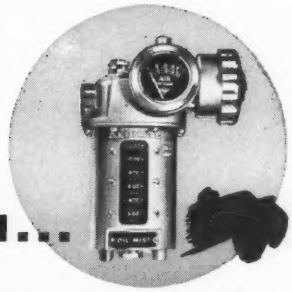

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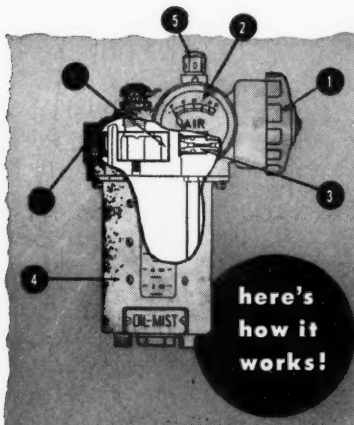

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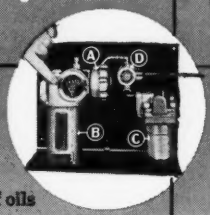

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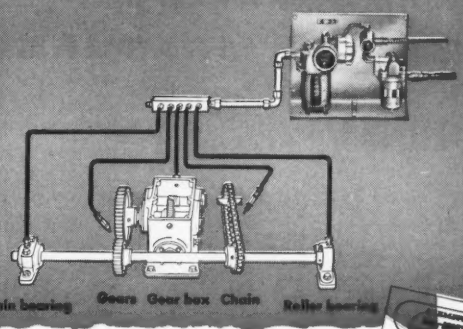

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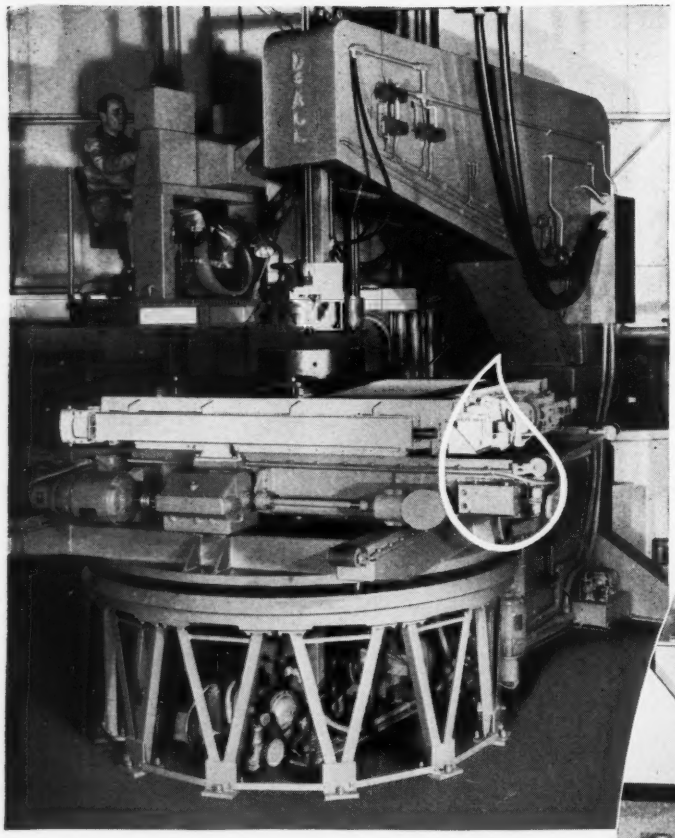

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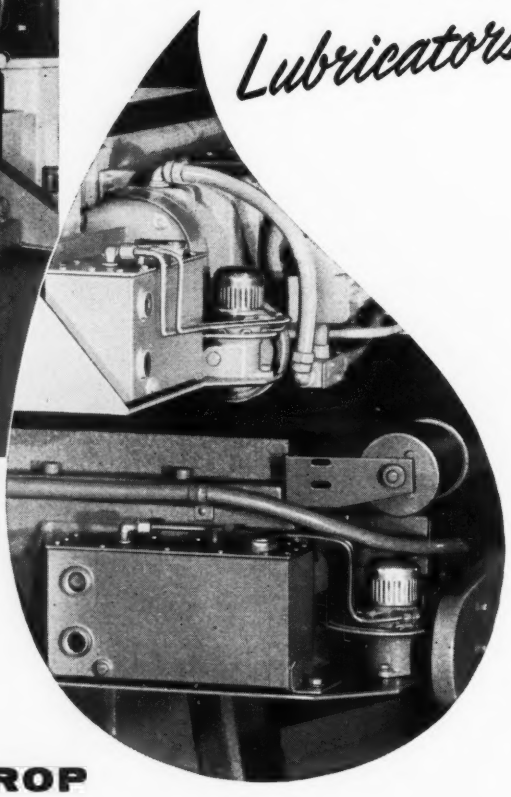

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Maintenance Of Soluble Oil Emulsions In Large Systems, by O. L. Maag & H. T. Peeples (The Timken Roller Bearing Co., Canton 6, Ohio).

The two largest plants of The Timken Roller Bearing Company are located in Canton and Columbus, Ohio. These two plants house hundreds of grinding machines which require, of course, large quantities of grinding coolant.

Scope of the Problem. At the Timken Company's Canton plant three different cooling systems circulate 100,000 gallons of soluble oil solution; one 40,000-gallon system, and two 30,000-gallon systems. The three cooling systems circulate continuously at the rate of 850 to 1250 gallons per minute. A seven-to-ten percent oil concentration is used. This proportion affords maximum rust protection of work and machines, has sufficient cooling properties, and possesses ample lubricating properties for guides and slides. Grinding wheels of suitable grit and grade have been adapted to the seven-to-ten percent oil concentration.

Grinding Mud Removed. Time cycle for settling dirt in the systems is approximately 25-30 minutes. Grinding mud is thoroughly flushed out of the machines into floor trenches. The mud is then drained from the trenches into a large sump, from whence it is pumped into deep metal tanks for settling. In one tank (Fig. 1) the mud settles to the bottom where it is carried out by means of a chain conveyor into a dump buggy (Fig. 2). The dump buggy hauls the mud out to a waste dump. As a result of this process, both the grinding wheels and the work pieces are adequately cooled by a clean soluble oil solution circulated by the three cooling systems.

Composition of the grinding mud varies considerably. It is comprised roughly of ten percent oil and organic compounds, forty percent metal, five to ten percent grinding wheel dust, and the balance water.

The settling tank at the Timken Company's Columbus plant is identical to the Canton plant's tanks, except that at the Columbus plant the tanks are cleaned with a scraper-type mud remover (Fig. 3). Efficiencies of the systems at Columbus and at Canton are about the same, although the Columbus cooling system is perhaps more rugged and more easily maintained. Also, the Canton plant uses a 1250-gallon-per-minute pump for its system, while the Columbus plant uses an 850-gallon-per-minute pump. Thus far these differences have proven to be insignificant.

Experiments have been conducted to determine whether additional filtering of the settled solution would improve the product enough to warrant a further investment in filtering equipment. Evidence thus far indicates no justification for further filtration of the coolant.

Aeration Stems Bacteria Growth. The oil coolant is pumped continuously and aerated constantly to avoid rank odor and rancidity. By diligently following the aeration principle it is not necessary, with stable soluble oil emulsions, to replace a coolant system completely. Practice at Timken is to analyze the solution daily for oil content and make the correct addition of both water and soluble oil. The correct level and strength of solution is thus kept in a system. This method does not eliminate bacterial growth. Warm weather naturally produces higher bacteria count.

(Continued on p. 11)

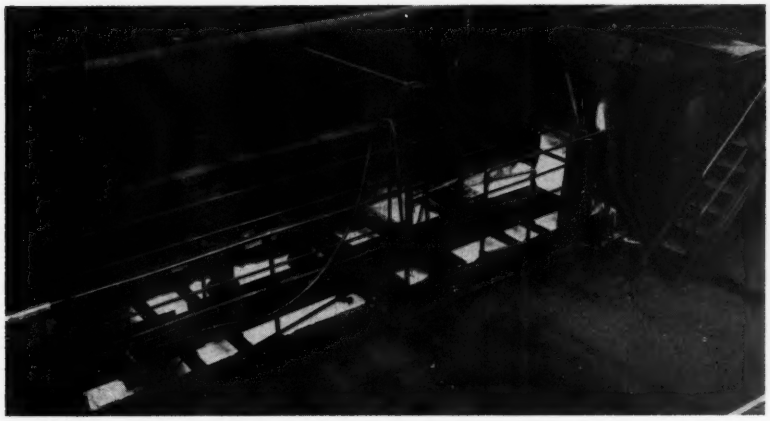

Fig. 1. Settling tank for soluble oil emulsions; Canton plant.

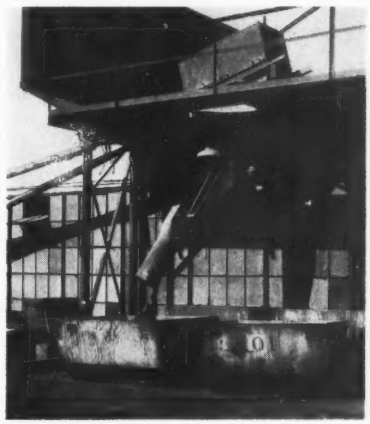

Fig. 2. Grinding dirt is emptied into dump buggies.

Fig. 3. Settling tank for soluble oil emulsions; Columbus plant.

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Low Temperature Factors . . . Does not harden, crack or decrease in adhesion.

Abrasive Resistance . . . Repellent to adhesion of scale, metallics and other contamination.

(Continued from p. 9)

Two Ways to Achieve a Sterile Solution. A sterile solution can be obtained by heating and circulating the solution at 150-160 degrees F. This process loosens dirt settled in pipe lines and helps to clean the piping system. However, the bacteria grow very rapidly after the solution cools to normal ambient temperature.

A sterile solution can also be obtained by adding 1 part formaldehyde (40%) per 1000 parts of solution. This method has been used when pumps are down for repairs or other emergencies, and over a week-end in hot weather, or during vacation shutdown. This treatment also keeps the solution from becoming rancid. The formaldehyde is gone in a few hours after the solution is again aerated, and, likewise, the bacterial growth is back to normal in a very short time.

Chlorine and other type bacteria killers have been tried out, but have been of no practical value.

Continuous aeration of good stable soluble oil emulsions has been the most satisfactory in keeping grinding solutions in good serviceable condition, and the scraper-type dirt remover has proved to be the most rugged and reliable.

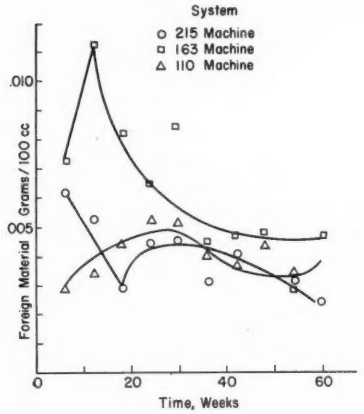

Fig. 4. Six-week averages of foreign material content of soluble oil grinding solution as it is being delivered to grinding machines in Canton plant recirculating systems; settling tank method of foreign material removal.

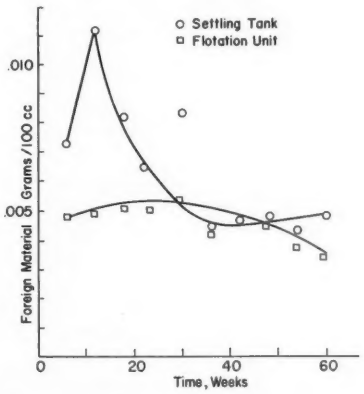

Fig. 5. Six-week averages of foreign material content of soluble oil grinding solutions, (1) settling tank method of foreign material removal; (2) flotation unit method of foreign material removal.

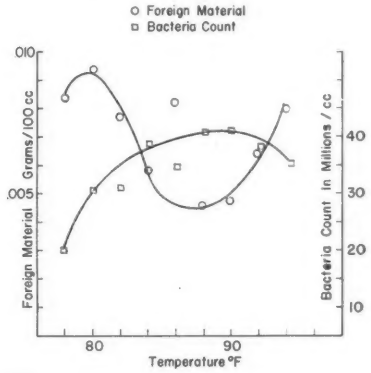

Fig. 6. Effects of temperature on foreign materials and bacteria content of soluble oil grinding solution.

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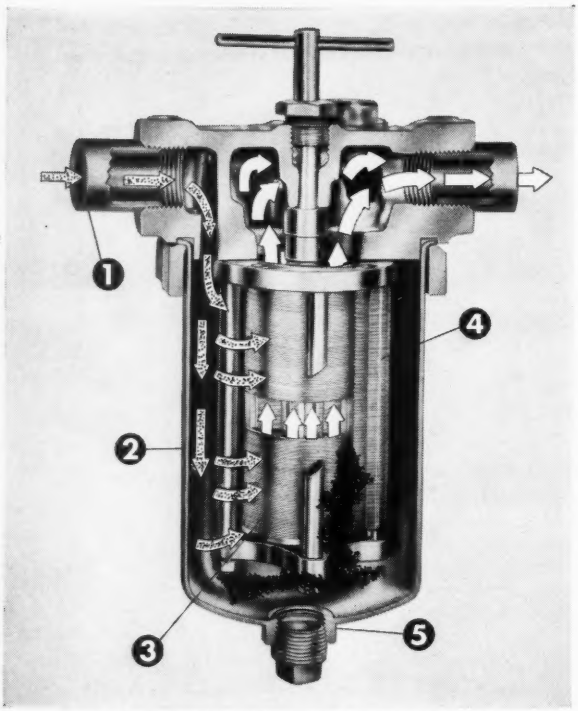

HOW IT WORKS. Dirty oil enters inlet (1) at left, fills housing (2) and flows through metal edge-type filter (3). Clean oil rises through center of filter, leaves at right. Dirt combed out by cleaner blades (4) is removed through drain (5).

Cuno's new SUPER Auto-Klean is the first practical, compact micronic filter for lubricating oil, hydraulic fluid, coolant, fuel, and other industrial fluids.

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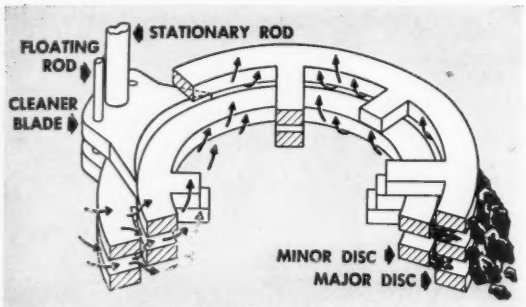
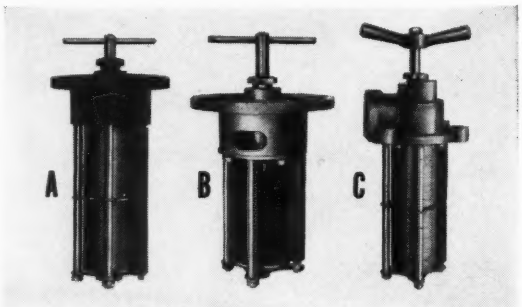
5. **High capacity in a small package.** Many times smaller than replaceable-cartridge type filters of equal capacity, it saves with lower initial costs, lower installation costs.
6. **Easy to build into new equipment.** Available for line-type housings *(left)* or incorporation in sump or reservoir *(below.)* Works in any position.
7. **Easy to install in old equipment.** Fits existing Auto-Klean housings. You can easily replace most 2¼" diameter elements with SUPER Auto-Klean for finer filtration.

Write today for complete technical data on the new SUPER Auto-Klean for your new or existing equipment. Ask for Catalog No. SAK-057. Cuno Engineering Corporation, 13-1 South Vine Street, Meriden, Connecticut. 5.9



FILTER ELEMENT consists of stacked major and minor discs and cleaner blade bearing against minor discs. Small arrows show flow. Short restrictions between minor and major discs stop 40-micron particles but allow high flow rate.



FILTERS FOR INTERNAL PIPING *(A and B above)* allow streamlined design plus the best infiltration. Flange mounting with external outlet *(C above)* and line-type *(large cutaway above left)* are just a few of many other possibilities.

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Lube Lines

by A. F. Brewer*

BENEFITS OF AUTOMATIC LUBRICATION. The extent to which benefits can result from employing means for automatic lubrication in machinery operation, depends upon the adaptability of the lubricating system to the machine and to the operating conditions. The net result can be most-pronounced when the lubricating system has been studied in connection with machine design. Then it is practicable to arrange for the required flow of oil or discharge of grease to bearings with due regard for the severity of the service, the importance of instant lubrication when the machine is being started, the extent to which the oil must act as a heat transfer agent, and the location of controls or metering devices. On an existing machine which is being converted to automatic lubrication, control valves, manifolds, and other fittings sometimes may have to be located rather inaccessibly due to presence of other piping or machine parts. The primary objective is to locate such piping for lubrication so that it will be as protected as possible against chance damage from tools, etc., which might cause leakage.

The function of any automatic lubrication system goes beyond the mere delivery of more-or-less measured amounts of lubricant. It is an adjunct to low maintenance costs which, normally, over a reasonable period of time will far outweigh the original cost of the lubricating system. Furthermore, it is an adjunct to protected lubrication because, when lubricants are delivered directly to circulating oil tanks or from bulk grease containers, there is far less chance of contamination from outside sources, i.e., moisture, dirt, plant dust, or lint.

Automatic lubrication, with the exception of a strictly splash-oiling system or pre-lubricated ball or roller bearing, implies that a certain amount of pressure is involved in conveying the lubricant to the operating parts. Automatic lubrication also suggests that a number of parts are being lubricated off the central system.

This matter of pressure is interesting. Sometimes, as with grease lubrication, it may be quite high. The system, of course, must be designed to withstand the maximum pressure which may be involved. For example, the nylon-type tubing used with a well-known grease lubrication system for automotive chassis is tested to a bursting pressure of at least 2500 psi.

Manufacturers' data on grease lubrication for heavy industrial use indicate that up to 96 points to be lubricated can be serviced by the master lubricator.

Pressure, especially in a grease-lubricated system, must be carefully regulated. If it is too high for certain types of bearing seals, grease may be forced past the latter and onto the exterior of the bearings, perhaps to contaminate goods such as textiles or foodstuffs which are being processed; or otherwise to drip or throw onto the floor to become a hazard to personnel. In an oil-lubricated system excessive pressure can cause aeration and foaming of the oil due to excessive splashing. Sometimes, also, as in a grease system, seals may leak to cause a sloppy hazardous condition around the machine.

There are, of course, quite a number of systems which develop automatic lubrication. They range from the simplest form, which is the waste pad which serves only one bearing, to the multiple-outlet centralized pressure systems. They are adapted to oils of a fairly-wide range of viscosity or to a normal consistency grease. (Note the accompanying Table.) In turn, oil lubrication may be fluid, or the system may convert the oil to mist by injection drop-by-drop into a stream of low-pressure air. Mist or spray lubrication has been widely accepted for high-speed precision mechanisms and, more recently, for gears, chains, and wire rope. This latter is an important development when long lines of cable or rope are to be lubricated, and where otherwise quite heavy-bodied lubricants would be used requiring heating and laborious handling.

AUTOMATIC LUBRICATION

Type	Typical Service	Oil Viscosity @ 100°F		Grease NLGI No.
		Sec. Saybolt	Centistokes	
Waste pad	Trunnion bearings in cement mills; textile mandrel bearings	300-800	65-176	—
Sight feed oilers	Engine, pump and compressor crosshead guides	250-550	54-121	—
Wick feed oilers	Line shaft bearings	250-400	54-88	—
Ring, chain or collar oilers	Plain bearing electric motors, pillow blocks	200-400	43-88	—
Gravity systems	Industrial steam engine crankcases, marine turbines	300-550	65-121	—
Mechanical force feed lubricators	Steam cylinders, horizontal compressors	* 300-550	* 65-121	—
Hydrostatic lubricators	Steam cylinders	*	*	—
Splash oiling	Engine and compressor crankcases	300-550	65-121	—
Pressure circulating systems	Steam turbines	140-350	30-77	—
Oil mist or oil fog systems	High speed bearings, plain or anti-friction; enclosed gears, chains, slides and ways	75-1000 (can be straight mineral or E. P. type oil. *Do not* use compounded oils)	14-220	—
Automatic metered lubrication	Enclosed bearings and gears on textile machinery, machine tools, bottling and food processing machinery and printing machines	100-800	21-176	—
Compression grease cups	Line shafting	—	—	1 or 2
Pressure gun application	Machine tool bearings, etc.	—	—	2
Centralized pressure grease systems	Ball or roller bearings on food handling or steel mill machinery, etc.	—	—	2

* Indicates use of steam cylinder oil of viscosity in accordance with operating conditions.

(Table reprinted from "Basic Lubrication Practice," by A. F. Brewer; Reinhold Publishing Corp., New York, 1955.)

LUBRICATION CHECK-UP PROCEDURE. There is a trend in industry to develop vital statistics with regard to machine life as related to lu-

(Continued on p. 70)

*Consultant, and Author of "Basic Lubrication Practice."

"Alloys for the Furnaces." Agnes Potter Lowrie depicts the ingredients which give steel its special qualities: reddish manganese for deeper hardening; ice-white quartz for flux, flanked (left) by tungsten and (right) by chrome, corrosion preventives for special-purpose steels; dusky lead glistening with molybdenum which maintains hardness at high temperatures; and (bottom row) samples of nickel, molybdenum and chrome, vanadium, copper.

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Advances in mechanics have given us *more* steel. Today's four-high mill, for instance, rolls more steel in a single shift than the old-time hand mill produced in a week.

Contributing mightily to more efficient production are equal improvements in the materials which smooth the process of steel making — Ironsides roll neck and roller bearing lubricants.

Roll neck shield, once applied by hand, is now supplied in liquid and semi-liquid form for continuous application by automatic pressure systems. Roll necks may be kept at constant temperature; the roller need not adjust his mill to maintain proper gauge.

More important, roll neck shield is formulated to the special needs of the individual user. For example, to compensate for local water and temperature conditions or to meet personal preferences of mill superintendents.

Roller bearing shield, an extreme pressure lubricant, has load-carrying capacity in excess of bearing manufacturers' standards, assuring a safety factor for peak loads. It carries the approval of all top bearing makers.
Today almost every major steel producer uses Ironsides lubricants. Our unique position is due to our ability to formulate for special applications and supply those formulas in any quantity.

We like tough problems; we've solved a lot of them. For example, Palmoshield, replacement for palm oil and most significant advance in lubrication since World War II. If you have a problem, we'd like to help lick it. A letter will summon one of our research engineers. The Ironsides Co., Columbus 16, Ohio.

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Men of Lubrication

by R. E. Archibald

NIKOLAY P. PETROV. Lubrication as a science may be fairly said to have started in the year 1883, for in that year the papers of Beauchamp Tower and N. P. Petrov appeared. Tower's work ultimately led to more far-reaching results; but it was, after all, only a well-executed experiment and would scarcely be remembered if it had not led directly to Osborne Reynolds' theory. Petrov, on the other hand, did a great deal of experimental work and gave a rational interpretation of it. It seems fair to say, therefore, that Petrov's work had more intrinsic value than Tower's.

Nikolay Pavlovitch Petrov was born in 1836. He was of the gentry of Novgorod Province. He attended the Constantine Artillery Academy and the Nicholas Engineering Academy, and later the Pedagogical Institute where he studied mechanics under the famous mathematician, M. V. Ostrogradsky. In 1865 he was sent abroad for further study in applied mechanics. In 1867, in recognition of his many accomplishments, he was named adjunct-professor in the Nicholas Engineering Academy. He held the chair of Steam Engineering at the Technological Institute, and from 1871 the chair of Railway Vehicles. In 1873 the Main Society of Russian Railways invited him to be a member of its engineers, and in 1876 the society appointed him as an expert or, as it was called in the United States, a "judge" to the World Exhibition in Philadelphia. From 1882 to 1892 he was President of the Temporary Council of the Directorate of State Railways, Director of the Department of Railways, and Chairman of the Engineering Council of the Ministry of Communications. From 1893 to 1900 he held the post of Freight Minister of the Ministry of Communications. Petrov appears to have lived to a great age, but it is uncertain when he died.

Petrov seems to have been dissatisfied by the work of his predecessors at the St. Petersburg Technological Institute, and he attempted to apply his own abilities to the independent solution of some important problems of the time. His first work, "The Outlining of the Cogs of Large Cylindrical Wheels by Means of the Arcs of Circles," was published in the "Engineering Journal" in 1870, and was awarded the prize of that journal. Following this he wrote significant papers on the wear of steel tires and on the dynamical loading of railroad rails (see Timoshenko: "History of Strength of Materials," McGraw-Hill, 1953). His work on lubrication was motivated by economic considerations. Petrov showed that the residues from the oil refining industry in Russia could be used as satisfactory substitutes for the animal and vegetable oils that were then the common lubricants. In the course of this work he deduced the value of the frictional couple on a shaft from its bearing due to the lubricant viscosity. He showed that the selection of a suitable petroleum oil for a lubricant depended, in the main, on a proper choice of viscosity. For this work he received the Lomonossoff prize of the Imperial Russian Academy of Sciences. In 1886 he contributed another paper on this same subject for which he was also honored by the Academy of Sciences.

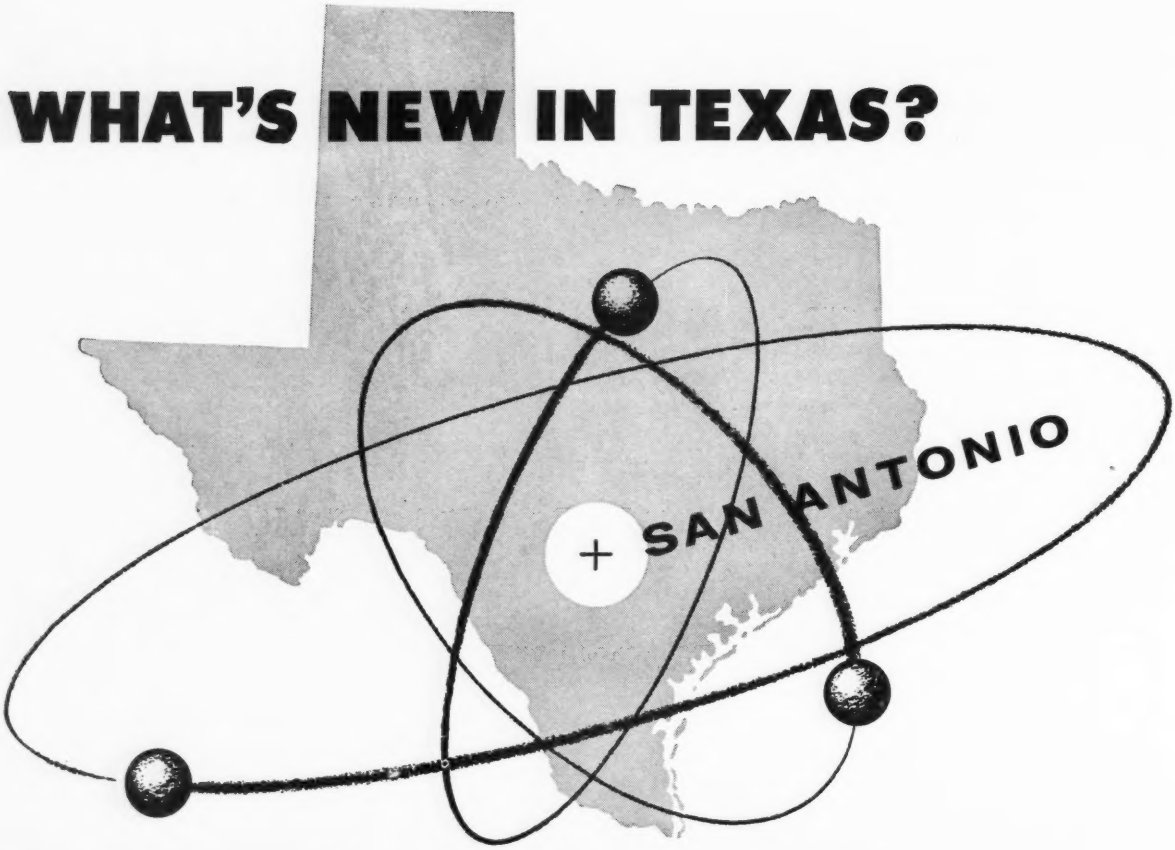
(Photograph courtesy of The Library of Congress)

The formula deduced by Petrov for the friction on a journal due to the viscosity of the lubricant, contains two terms for the slip of the lubricant at the boundaries. When these terms, which are now known to be unnecessary, are omitted, the formula reduces to the well-known one for the cases of two concentric cylinders having a relative motion of rotation. This formula is not accurate for the case of a heavily-loaded journal bearing where the journal eccentricity has a marked effect. But for lightly-loaded bearings it is a good approximation to the frictional torque, and the formula is very useful.

The problem of the torque between a pair of concentric cylinders having a viscous fluid in the annular space, and having a relative motion of rotation, has an interesting history. Sir Isaac Newton considered the problem in his discussion of viscosity (Principia, Proposition LI Theorem XXXIX Cor. II), but his solution is in error. The exact solution was first given by Sir G. G. Stokes in 1845. As Stokes points out: Newton made the error of assuming equal forces on the inner and outer surfaces of any fluid annulus. The condition for uniform motion is, rather, that the moments of these forces about the axis be equal. However, in the case of a bearing where the film is thin the refinement due to Stokes is not material, and the approximate theory

(Continued on p. 72)

WHAT'S NEW IN TEXAS?



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LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Volume 12 No. 1
January-February 1956

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J. W. Hopkinson, President
American Society of Lubrication Engineers

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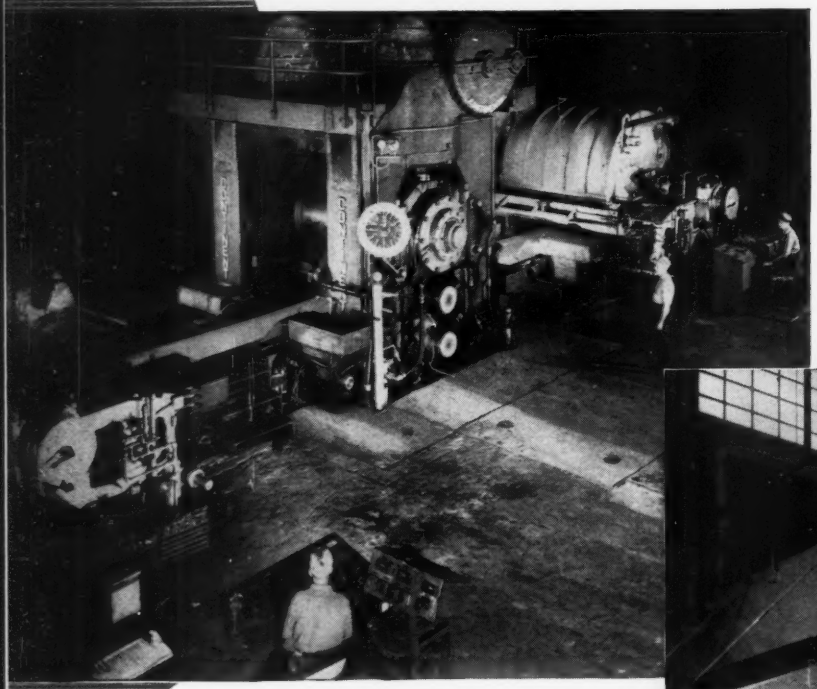
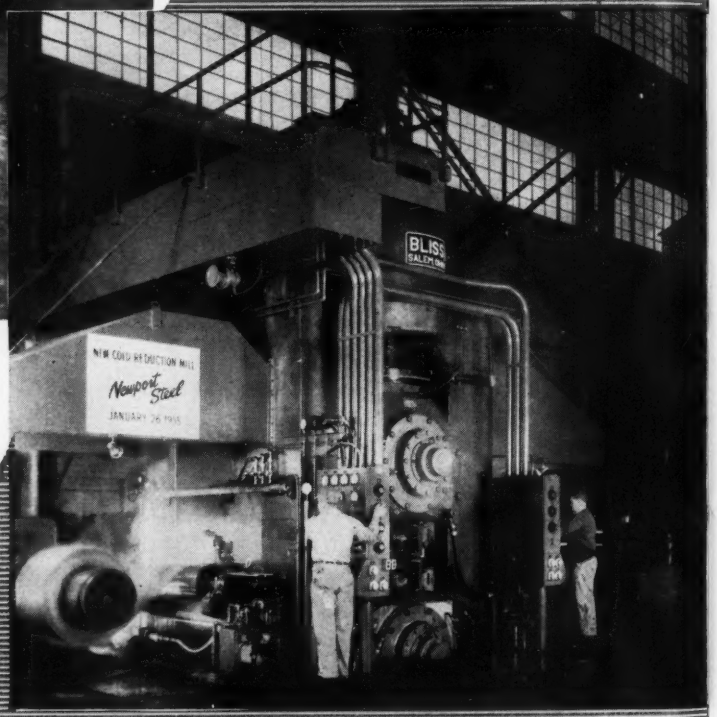
HOT or COLD





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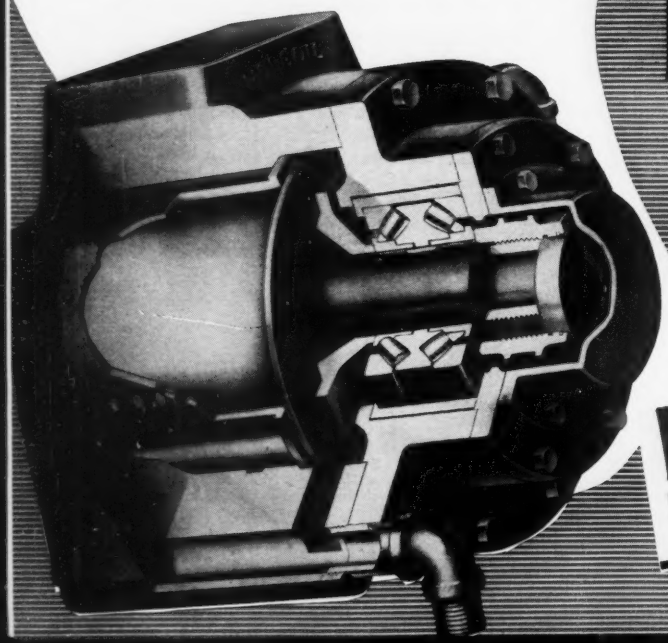

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PRACTICAL LUBRICATION, No. 17

Protective Oil Coatings

by K. F. Schiermeier & R. W. Lewis*

The complicated task of preserving industrial, agricultural, and military equipment has necessitated the development of several types of rust preventive compounds. These compounds are classified and discussed here under four headings: the rust preventive oils, water displacement oils, semirigid soft coatings, and hard tough coatings. The factors which influence the selection of one of these classes for a specific application are considered. Some of the more important basic factors which must be considered in the formulation of rust preventives are presented along with a discussion of the relationship between chemical structure and effectiveness of rust preventive additives.

The principle of protecting surfaces which are susceptible to decay, rusting, and corrosion by the application of a relatively inert film has been known for many centuries. Liquid or semiliquid coatings which can be placed in the protective oil class have also been used for a long time. The first materials of this type were animal oils and fats such as tallow, lard oil, fish oils, and sperm oil. Some of these were excellent rust preventives and are still widely used.

Until the late 1930's little further important progress had been made in the protective oils field. Uninhibited mineral oils were used in fairly large volumes to prevent corrosion, and mixtures of mineral oils and the above fatty oils were employed where the rusting conditions were severe. Undoubtedly the greatest impetus to product development and fundamental studies in this field was the advent of World War II. In this conflict enormous quantities of material, much of it consisting of precision instruments which depended upon smooth, closely fitting parts for satisfactory operation, were shipped, stored, and used. A basic weapon, the rifle, is a good example. In World War I and early in World War II the bolt action Springfield was used. This was a sturdy, dependable weapon relatively insusceptible to jamming by mud, sand, or rust. It was replaced by the semiautomatic Garand, a more effective weapon but much more intricate and susceptible to the above contaminants including rust. Add to this the fact that these instruments were subjected to some of the most extreme conditions of exposure that can be found; for example, the hot humid atmosphere of the jungle and the salt laden atmosphere of the beach and ship deck. With large percentages of the equipment reaching the point of operations in unusable condition, the demand by the services for effective rust preventives became urgent. As a result industrial, university, and government laboratories carried out extensive research programs in the field of rust preventives.

Types of Protective Films, and Factors Influencing Selection of Proper Type. As a result of the concentrated effort during the war and the years since, there now exists a number of specialized types of petroleum-derived rust preventives. There are several methods of classifying these types; the breakdown used here will emphasize the physical nature of the coating itself. Before discussing the various kinds of products available, however, it is desirable to note the reasons why different types of rust preventives are required. Factors which influence the selection of a suitable compound for a given job may be listed as:

(A) Degree of protection required. This may vary from a few hours on a shelf indoors to a few years outdoors in unprotected storage.

(B) Physical properties of the film. Situations are not unusual where odor, color, melting point, hardness, toughness, viscosity, volatility, and other characteristics may be very important.

(C) Mode of application desired. These include spraying, brushing, swabbing, dipping, and flowing on.

(D) Nature of the surface to be protected. Whether the surface is wet or dry is a very important consideration. The presence of nonferrous metals such as magnesium, lead, zinc, and others which may be susceptible to chemical attack by the preservative is often important.

(E) Ease of removability desired. In some applications it is necessary to remove the protective coating before the component is put into service. It is often true that ease of removal must be sacrificed to obtain better protection, or vice versa. Selection of the optimum coating may necessitate a compromise.

With these factors in mind, the various classes of rust preventive coatings can be discussed.

(1) Rust preventive oils. The relatively low viscosity oil coatings are the most versatile and widely used. This type of coating is the principal topic of this discussion. If it were possible to achieve adequate protection with oils, they would largely replace most of the other types. They are easy to apply and remove; the films are clear, odorless, clean to handle, and are self-mending if ruptured. Unfortunately, the degree of protection obtainable with oil films is limited; straight, refined mineral oils give satisfactory performance only for very short periods under high humidity conditions. If one of a number of types of addition agents is incorporated into a mineral oil, the protection achieved can be increased appreciably. These addition agents include the organic fatty acids, their salts, and their esters, various salts of petroleum sulfonic acids, esters of phosphoric acid, and amine salts of carboxylic acids. Oils can be prepared containing one or more of these additives which will prevent rust-

*Shell Oil Co., Wood River, Ill.

ing for as long as 6-12 months in protected storage. To date, however, preservative oils have not been prepared which are recommended for outdoor storage. It is of interest to point out that the refined mineral lubricating oils are excellent carriers for the effective rust preventive addition agents. They are relatively cheap, non-volatile, moderately good solvents, and have good lubricating properties where needed. There are several other satisfactory carriers or base materials such as the high molecular weight esters, polymeric ethers, alcohols, and olefin polymers. However, the relatively high price of these synthetic materials limits their use to highly specialized jobs.

(2) Water displacing rust preventive oils, fingerprint removers. The water displacing oil is intended for use where the ferrous surface to be protected is in contact with water. A very common application, and one which uses large volumes of this type of product, is the protection for a short period indoors of steel and cast iron parts which have just been machined with a water-emulsion cutting fluid. Often the water-displacing oil will be used only to "dry" the parts and to protect them until a more permanent rust preventive film can be applied.

Fingerprint removers are similar in composition to water-displacing oils, except that the latter contain little water and are intended to pick up or dissolve water while fingerprint removers initially contain the water necessary to dissolve organic and inorganic fingerprint deposits. Fingerprint removers are usually used on finely machined equipment after intermediate manufacturing steps or prior to packaging.

Compounds of the above type contain low-viscosity oils or rather viscous oils plus a volatile solvent. Blended with the mineral oil are usually several surface-active additives.

(3) Semirigid, soft, waxy coatings. Films having these characteristics may be obtained either with products composed of a volatile solvent plus a nonvolatile base material, or with products consisting of nonvolatile components only. The former type may be a liquid which can be applied at room temperature by spraying or brushing. The products composed of only nonvolatile bases usually have a paste-like consistency and must be applied hot by dipping, swabbing, or brushing.

It will suffice here to mention a few typical components of this class of rust preventive. The solvent diluent is usually mineral spirits, kerosene or, when greater solvent power is needed, the aromatic fractions of these diluents which can be obtained by SO_2 extraction. In special cases nonflammable solvents such as trichloroethylene are used. The nonvolatile component may be wool grease or its derivatives, low melting point (ca. 130-150 F.) petroleum waxes or, less frequently, vacuum-reduced asphalt. Viscous refined lubricating oils are often included along with rust preventive additives. Water displacing characteristics can be built into this class of rust preventives also.

The important advantage of this type of rust preventive film, compared to the previous oil coatings, is the much greater protection which can be achieved. Products are marketed which afford heavy films and which will prevent rusting as long as two years in outdoor unprotected storage and indefinitely in indoor or shed storage. Compared to the low viscosity oils they suffer a number of disadvantages, however. The hot dip compounds are more tedious to apply and the volatile solvent compounds may create fire hazards (nonflammable solvents are usually toxic). The film is tacky and, hence, more disagreeable to handle. Likewise they are somewhat more difficult to remove. These films are also easy to rupture by rough handling but, fortunately, usually possess some self-mending characteristics, to a somewhat lesser degree than do the oils.

(4) Hard, tough coatings. These important rust preventives most closely approach the paints in physical characteristics and protective properties, and are mentioned here because they often contain large proportions of petroleum derived materials. The coatings differ from paints in being removable (easily removable compared to paint, but very difficult compared to the above oil coatings); they have little or no decorative properties but are usually much less expensive than paint. Products which give this type of film consist of nonvolatile bases dissolved or suspended in volatile solvents. The solvents are of the paint-thinner type, mineral spirits, or aromatic extracts. The nonvolatile portion is often a special, high melting point, high penetration index (tough, elastic) asphalt. If a clear coating is desired, however, most or all of the asphalt must be replaced with a material such as ester gum, refined petroleum resins, or one of the synthetic polymers used in varnish manufacture. Surface-active additives are sometimes used in these products but are less important than in oils. Protection is due principally to the mechanically impervious film.

The outstanding advantage of this class of preservative is the excellent protection afforded along with a tough, rupture-resistant, nontacky film. The clear-coats generally give appreciably less protection than the asphaltic coatings, but they do make it possible to see the surface underneath, read serial number, and inspect for rusting without removing the film. The principal disadvantage of the hard coating is the difficulty encountered in removing the film. A reasonably good, preferably aromatic, solvent such as a light distillate fuel oil, together with some sort of scrubbing action is usually required. These materials are often used as inexpensive paint substitutes where a long lasting, nondecorative film is required.

Basic Considerations in the Formulation of Rust Preventives. The previous section makes it clear that there are good reasons for the existence of a number of different types of rust preventives. It is usually not difficult for the lubrication engineer or, for that matter, a farmer or home owner to define the basic requirements which a product should meet. When the chemist sets out to meet

these basic requirements, there are a number of factors which he must consider. These can probably best be discussed in the light of a specific requirement. For example, a general purpose rust preventive lubricating oil might be needed to meet the specifications shown in Table I.

Table I.

Viscosity at 100 F., SSU	350-400
Viscosity at 210 F., SSU	52-58
Properties of Coating	
Sheltered Outdoor, Months	3 min.
Indoor, Months	6 min.
Unsheltered Outdoor	Not recommended
Humidity Cabinet, Hours	300
Salt Spray, Hours	48

Products similar to this are widely used in industry for preserving parts and equipment in sheltered storage, for preventing rust on steel sheet enroute to the fabricator, and for lubricating and preserving various types of operating machines; it is a good "oil can" lubricant for the amateur carpenter and mechanic.

The first step in developing such a product in the laboratory is to select the basic component, the petroleum oil. This is a simple task since most petroleum refiners make lubricating oils which bracket the viscosity required. For this product the oil may be distilled and refined from either a wax or asphalt base crude. The selection of the component(s) to give the oil the rust preventive properties required is not so simple; the base oil would prevent rusting for only a few days to a month in sheltered outdoor exposure, depending on climatic conditions, and less than 24 hours in the humidity cabinet. Increasing this protection time to three months or more in sheltered outdoor exposure or to 300 hours in the humidity cabinet is a rather difficult task with an oil film which after draining is only 0.0006 of an inch thick.

The problem, then, is to add some material to the oil which will greatly enhance the protective characteristics. It is well known that rusting of iron results largely from an electrochemical reaction involving water and oxygen as well as the metal. The end products may be FeO which is light gray or Fe_3O_4 which is black (and is sometimes mistaken for organic lacquers or stains) or Fe_2O_3 which is red. Prevention of rusting by exclusion of oxygen is impractical since oxygen is appreciably soluble in hydrocarbons as well as other organic liquids. However, effective protection can be obtained if access of moisture is prevented. Mineral oils do not form an effective barrier to moisture but certain polar organic compounds, i.e., molecules having a dipole, when added to mineral oil do form fairly impervious barriers. The polar molecules which are effective are those containing hydrocarbon-soluble oleophilic portions, such as long esters, phosphates, or sulfonates. There is little doubt that these compounds function by being adsorbed on the metal surface, being attached in an oriented fashion to the metal by their polar "heads" and having their oil soluble "tails" extending into the oil. The oriented films then hold the carrier oil in their interstices, and the combination of oil and additive prevents entrance of moisture to the steel.

Experiments which strongly support the above mechanism for rust prevention by oil additives have been carried out.[1] In one case iron powder was contacted with samples of white (medicinal) oil containing various classes of organic compounds. The powder was filtered off and the amount of additive removed by the powder was determined. Three types of adsorption isotherms were obtained in these experiments. They are shown in Fig. 1. All of the additives that had isotherms of Types 1 and 2 were good rust inhibitors while most that gave Type 3 were poor. Since the surface area of the iron powder was known (by nitrogen adsorption measurements), it could be calculated that the poor Type 3 additives were adsorbed to a thickness of only 1 to 2 monomolecular layers. Types 1 and 2 gave films several molecular layers thick. The tabulation appearing below Fig. 1 gives examples of the Types 1, 2, and 3 additives.

In another type of experiment, films of rust preventive additives were built up on polished steel surfaces by the Blodgett technique. An oriented monomolecular film of the additive was first spread on distilled water and then transferred to the steel by lowering the polished panel in a vertical position

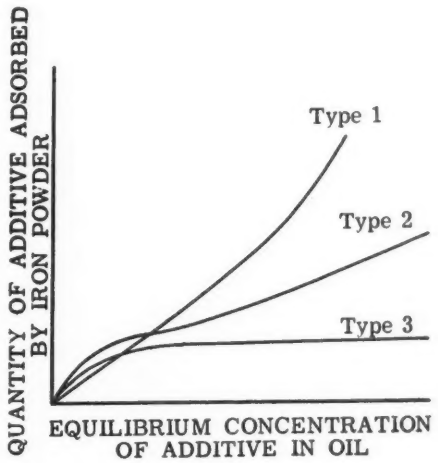
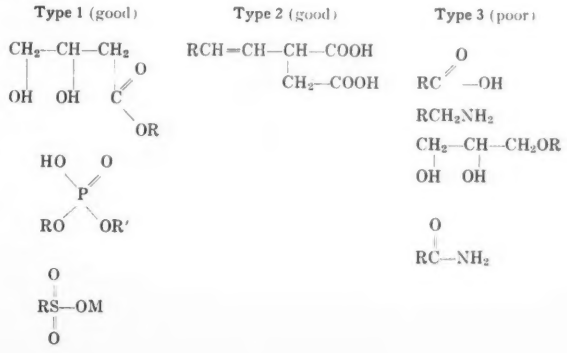


Fig. 1. Adsorption isotherms of rust preventive oils. (Below) R = long alkyl chain; R′ = short alkyl chain; M = alkaline or alkaline earth metal.



beneath the water surface and withdrawing it. A constant surface pressure was maintained on the additive film while the panel was being immersed by the use of castor oil which was separated from the additive film by a waxed thread. The most interesting observation from these experiments was the fact that none of the additives were effective in preventing rust regardless of the number of molecular layers built up. However, when the panels containing the layers of adsorbed additives were dipped in oil, considerable improvement in protective properties was noted for some of the additives. Even for the effective types it was noted that at least six molecular layers had to be present to prevent rusting. No protection whatever was obtained by one or two molecular layers of any of the additives tried. These experiments demonstrated then that adsorbed films several molecules thick are needed for good rust prevention, and that oil is not merely a carrier for the rust preventive additive. This latter finding helps account for the fact that the mineral oil type has frequently been found to influence significantly the protective properties of the finished rust preventive oil.

The most effective rust preventive additives, then, contain a very strong polar group, e.g., sulfonates or phosphates, or two or more less polar groups, i.e., hydroxyl, ester, carboxylate, mercaptan, etc. The latter class is much more effective if the polar groups are close together. The rust preventive effectiveness of these compounds is primarily a function of the balance between two forces: (1) The force of attraction between the additive and the metal surface; (2) The force of attraction between additive and the oil carrier.

A strong attraction for a metal surface is a prerequisite, but if an additive is too polar it may not have sufficient affinity for oil. (It may not be sufficiently oil soluble to permit the formulation of solution stable products.) An additive also may be strongly polar but attached to one or more long hydrocarbon chains which have a stronger affinity for oil than the polar group has for the metal. Such an additive will be too oil-soluble to effectively maintain an adsorbed film on the metal. Thus, an effective rust preventive is a compromise balance between these two extremes. In cases where an excessive amount of water comes in contact with the protective film, the force of attraction between the additive and water becomes important. If the additive is too polar, this attraction may be so great that the water leaches the additive from the oil film rendering the film ineffective.

With some additives, which are relatively insoluble in the oil base, it is necessary to add a solubilizing or dispersing agent. (This is usually a material which is mutually soluble with both the rust preventive additive and the oil.) In this way an effective concentration of a slightly soluble additive can often be held in solution or colloidal suspension.

In the discussion above it should be kept in mind, however, that some addition agents function by chemically combining with the metal surface and thus forming a protective layer of more inert reaction products. The strongly acid inhibitors, for example, are capable of forming protective iron soaps.

With this understanding of the mechanism of rust prevention and the relation between molecular structure and additive effectiveness, it is possible to select a few specific organic compounds to combine with petroleum oil in an effort to meet the detailed specification outlined in Table I. Some experiments of a "cut and try" nature are invariably necessary to select the specific compound and the correct concentration. To meet this specific requirement a polar compound having three moderately strong polar groups arranged close together, namely, glyceryl monooleate, was selected. A concentration of 4.5% was required to give the necessary protection. Since this quantity exceeded the solubility of the compound in petroleum oil, 0.5% of petroleum sulfonates were added to give a stable suspension of the undissolved additive. The resulting product had the desired physical properties and exceeded the protective properties required, as shown in Table II.

Table II. Protective Properties of Coating.

Sheltered Outdoor, Months	4-8
Indoor, Months	8-16
Humidity Cabinet, Hours	400
Salt Spray, Hours	48

There are numerous other types of rust preventive oils. The above basic considerations apply to each, but each must also be made differently in many respects to fulfill various requirements. For example, preservative engine oil additives combine rust prevention and detergency. Oxidation inhibitors are required in these oils as well as in a number of others such as gun oils, gas turbine lubricants, and instrument oils. The latter must often be specially refined from selected stocks to meet stringent volatility and low temperature viscosity requirements. Synthetic base oils, in certain instances, are also required.

In water-displacement or fingerprint remover oils the use of polar or surface-active additives is most important. Greater affinity for both a metal surface and for water is required than in other rust preventives. These additives, however, are less soluble in oil and other solubilizing additives, such as the alcohols or glycol-ethers, are often required for solution stability.

In the field of waxy and asphaltic coatings the degree of protection largely depends upon the nonpermeability of the coating to moisture and vapor which, in turn, mainly depends upon the thickness and structure of the coating. In general, addition agents do not contribute much to the protective properties unless they improve the smoothness and continuity of the film. A grainy, rough film is a less effective protective barrier. This sometimes leads to difficult problems in making wax-volatile solvent blends, wax being difficult to disperse as a colloid in even the aromatic solvents. If the wax is poorly dispersed, a grainy discontinuous film results when

the solvent evaporates. Crude waxes containing high molecular weight resinous materials or partially oxidized waxes are often more readily dispersed than are the purified petroleum waxes. With hot dip wax coatings the problem of grainy, discontinuous films is not so serious. With the asphaltic coatings the most serious formulation problems usually involve meeting the stringent physical requirements for the dried film, such as resistance to weathering, flexibility, hardness, high melting point, and dryness (nontackiness). Protective characteristics are seldom limiting if the coating has good weathering characteristics. Physical durability to resist rough handling is important. Detailed factors influencing the selection of asphalts for various applications are beyond the scope of this discussion.

In conclusion, it might be pointed out that the field of rust preventives, although still in its infancy, is one in which considerable progress, both of a fundamental and practical nature, has been made. This is a good example of the results which can be accomplished through the cooperative efforts of the industrial, academic, and government researchers when there is a common and urgent need.

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Section News

Notice to ASLE Section Secretaries: To insure early and adequate coverage of your Section's activities in **Lubrication Engineering**, all "Section News" copy (including Preprints or Abstracts of papers presented, Election of Officers, Photographs properly identified, etc.) should be received by the ASLE National Office 1½ months prior to the month of publication.

BOSTON, October '55 meeting. A. E. Baker, Manager of the Chemistry & Insulation Section of the Thomson Laboratory of General Electric Co., presented a paper entitled "Practical Methods of Evaluating Oils," covering some chemical and physical properties of cutting fluids, and how they have been studied in a practical way to determine their importance in metal cutting problems.

1956 Program Schedule: **January 16,** "Metal Forming Lubricants;" **March 19,** "Manfacture of Lubricating Oils;" **April 24,** Plant Tour of the Brown & Sharpe Mfg. Co., Providence, R. I., followed by a paper entitled "Machine Tool Maintenance;" **May 21,** Past Chairmen's Night. (Submitted by G. T. Collatz, Sec'y.)

BUFFALO, September '55 meeting. J. L. Zakin, Socony-Mobil Oil Co., presented a paper entitled "The Apparent Viscosity of Greases & Its Practical Aspects."

October. "Silicone Lubricants —Advantages & Disadvantages," by J. Wilson, Dow Corning Corp., followed by a color film entitled "What's A Silicone?"

January '56. Tenth Anniversary Meeting of the ASLE Buffalo Section.

Lubrication Engineering Course. The Buffalo Section is co-sponsoring with the Millard Filmore College of the University of Buffalo an evening course in Lubrication Engineering. First class meeting was held January 31, and two hour sessions will meet every Tuesday through April 3. Complete details can be secured from L. E. Locke, Battenfeld Grease & Oil Corp., N. Tonawanda, N. Y. (Submitted by L. E. Locke, Sec'y.)

CHICAGO, November '55 meeting. K. Hyatt, E. I. duPont deNemours & Co., presented a paper entitled "Motor Oil Additives." (Submitted by J. N. Waddell, Sec'y-Treas.)

FORT WAYNE, November '55 meeting. M. Steckley, Personnel Director for the Toledo Oil Refinery of Sun Oil Co., previewed a film entitled "The Magic Barrel" describing the many products made from a barrel of crude oil. (Submitted by J. W. Buckner, Chrmn.)

NEW YORK, December '55 meeting. E. R. Keast, Lubrication Engineer & Consultant on processing problems for Wright Aeronautical Corp., presented a paper entitled "Carelessness Costs You Money," discussing lubrication procedures used in the storage and handling of various pieces of equipment, and explaining the methods of training belt men and oilers. On the same program W. N. Nicholson, Vice-President & General Manager of the New York Div. of Alemite Corp., previewed a film entitled "Barrel to Bearing," followed by a discussion of the various methods for dispensing lubricants. (Submitted by E. Landau, Publicity Chrmn.)

N. CALIFORNIA, November '55 meeting. J. T. Wilson, Chief Physicist for The Louis Allis Co., presented a paper entitled "The Effects of Atmospheric Contaminants Upon Grease & Oil Lubricating Systems," illustrated with photomicrographs of lubricant contaminants.

December. "A Review of Some Hydrodynamic Lubrication Applications," by Dr. E. K. Gatcombe, Professor of Mechanical Engineering at the U. S. Naval Postgraduate School, Monterey, Calif. (Submitted by A. C. West, Sec'y.)

PITTSBURGH, November '55 meeting. C. J. Boner, Chief Research Chemist for the Battenfeld Grease & Oil Corp., presented a paper entitled "Application of Lubricating Greases," covering the types of greases in use today, their various properties, and their applications in industry. (Submitted by S. W. Settle, Publicity Chrmn.)

TWIN CITIES, October '55 meeting. H. L. Hemmingway, The Pure Oil Co., presented a paper entitled "What Is Going

(Continued on p. 60)

Wear Studies With Radioactive Gears

by V. N. Borsoff*

Test equipment and methods used to detect and measure gear wear by radioactive techniques are described. Tests performed with unreactive mineral and synthetic oils, and also with lubricants containing extreme pressure additives, show that three main types of wear exist in gears, namely: wear by scoring, wear by abrasion, and wear by chemical corrosion. These three types of wear are defined and discussed.

The wear tests were supplemented with metal transfer studies which showed that metal transfer invariably occurs under scoring conditions but does not necessarily occur under conditions leading to light abrasion or chemical corrosion. The significance of metal transfer with respect to the mechanisms of lubrication and wear are discussed.

In a long range research program on the fundamentals of gear lubrication carried out at Shell Development Company, the studies of wear characteristics of lubricants were made with the aid of radioactive tracer techniques. In conventional studies of gear lubrication, wear is measured by weighing or measuring the gears. The presence or absence of a thick film had to be inferred from the appearance of the contacting surfaces, and the score load was determined by visual detection of the incipient score marks. These conventional methods, besides being time consuming, also introduced some limitations. In measuring wear, for example, only the total wear for a given period could be obtained, but not the rate of wear at various times during that period. Removing the gear from the shaft for weighing and then reinstalling it introduced uncertainty in the alignment. All these limitations and uncertainties are eliminated with radioactive tracer techniques. In actual practice these techniques proved extremely valuable, not only because they permitted accurate and continuous evaluation of wear, but also because they supplied additional information useful in explaining the mechanism of gear lubrication. These features of radioactive tracer techniques, together with the data on the wear behavior of various lubricants, are presented and discussed in this paper.

Brief Description of the Technique Used. The technique consisted of irradiating one gear and operating it under controlled conditions with an inert gear. Two phenomena are usually studied simultaneously: wear and metal transfer. Wear is measured by placing a Geiger tube (or a scintillation counter) in the oil stream, observing the counting rate, and calculating therefrom the amount of metal removed from the irradiated gear. Metal transfer from the radioactive to the inert gear is determined by periodic monitoring of the working surface of the inert gear and by radiographing it.

The controlled operation of gears is secured by running them in our Spur Gear Machine shown in Fig. 1. This machine, based on the four-square or closed power circuit principle, permits loading the gears up to 6000 pounds per inch of face, and can be operated between 1000 and 5000 rpm. The gears used are 17-19 tooth, 6 diametral pitch, 20° pressure angle and 0.25 inch face width. These gears are hardened to 62 Rc and finished to 20 microinches. The 17 tooth gears were irradiated by neutron bombardment in an atomic pile at Oak Ridge, Tennessee. A description of the reactions taking place in the atomic pile and calculations of the resultant radioactivity were given in one of our previous presentations.[1] The test gear box was provided with a separate oil circulating system, shown in Fig. 2, which included a vessel "A" containing a Geiger-Muller tube, model TG-C4, connected electrically to a Berkeley Decimal Geiger-Muller scaler, model 1000. The vessel "A" was placed at some distance from the test gear compartment and also was shielded by thick lead-brick walls.

The tests consisted of operating the gears at 3000 rpm and increasing the load by 360 pounds per inch steps, running 30 minutes at each load setting. The counting rate within the oil stream was recorded every minute and converted to wear in milligrams of iron. For evaluation of metal transfer, the inert gear was removed, cleaned, and radiographed.

Test Results. The results of the first series of experiments are presented in Figs. 3 & 4. Two types of wear rates were observed: an instantaneous or very rapid rate, and a slow rate continuous over a prolonged period of time. Microscopic examinations of the working surfaces showed that the instantaneous increases in wear took place when scoring occurred. This happened with the unreactive

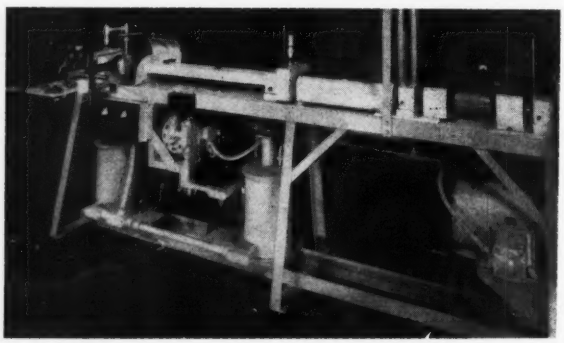

Fig. 1. Side view of gear lubricants testing machine.

*Shell Development Co., 4560 Horton St., Emeryville 8, Calif.

(This paper was co-sponsored by the ASLE Technical Committees on Bearings & Bearing Lubrication, and Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 13, 1955.)

mineral oils shown in Fig. 3. The graphs also show that the lightest oil gave the highest wear, and with it wear began at the lowest load. The other oils follow in order of their viscosity. At high loads wear became progressive, and as the microscopic examinations revealed, scoring and abrasion became intermixed.

With an E.P. compounded oil no wear was detected up to 50 pounds beam load (3600 pounds per inch). At higher loads wear became progressive and, since no scoring occurred, was attributed to abrasion or chemical corrosion, Fig. 4.

From these and other similar data, we have postulated that hard gears wear by three mechanisms: scoring, abrasion, and chemical corrosion. Wear by scoring occurs when the working surfaces are no longer separated by a lubricating film and direct metal to metal contact takes place. At light loads this wear is instantaneous in nature, and as soon as the high spots are removed the lubricating film reestablishes itself. The second type of wear is caused by abrasion, i.e., removal of metal by mechanical action. When gears operate with thick film lubrication, abrasion, as a rule, does not exist except for an occasional localized scratch. In the boundary zone of lubrication, as with E.P. compounded oils at loads above the score load of the base oil, wear by abrasion may become of importance. The third type of wear is caused by chemical corrosion. Theoretically, it may occur both in the thick film and boundary zones of lubrication. Under light operating conditions, when temperatures are low, chemical wear is small; but in prolonged operation it may become significant. These experiments also showed that wear by abrasion and corrosion is small, and for its effective study in relatively short periods of time the sensitivity of wear

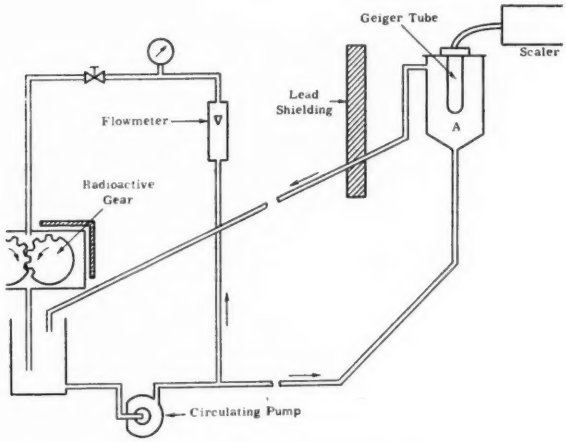

Fig. 2. Schematic diagram of gear-wear test apparatus.

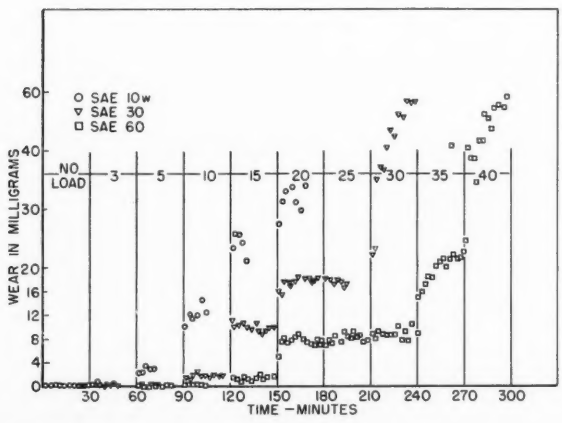

Fig. 3. Wear vs time and load; radioactive tracer technique.

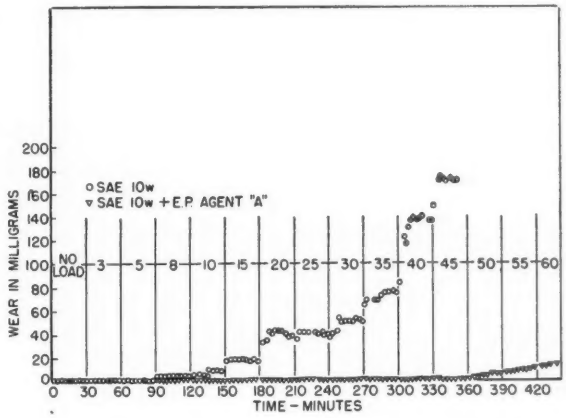

Fig. 4. Wear vs time and load; radioactive tracer technique.

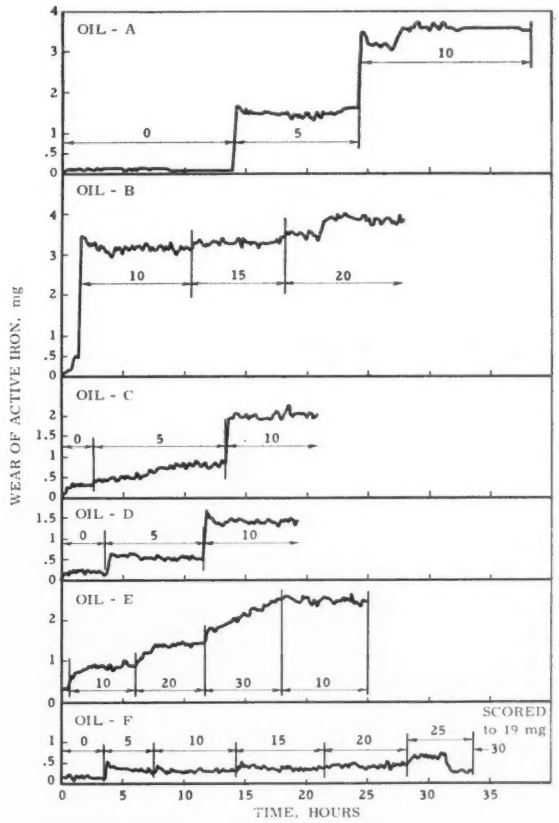

Fig. 5. Radioactive gear wear tests; mineral and synthetic oils.

detection must be increased.

Wear Studies with Scintillation Counter. To achieve the required sensitivity, the apparatus was modified by replacing the Geiger-Muller tube with a scintillation counter, Model SC-2A, manufactured by the Nuclear Research Corporation of Philadelphia. This counter was coupled electrically with a Precision Ratemeter, Model SC-34 of Tracerlab, Inc., and is also connected to a Recording Brown Instrument which traces the wear continuously. The oil circulating system was also modified to increase the efficiency for detection of radioactivity, and to make it possible to use smaller quantities of the test oil. The vessel "A" of Fig. 2 was made spherical and the sump abandoned. With these improvements the sensitivity of wear detection was increased to 0.01 milligram of iron. In the subsequent experiments the test procedure and the operating conditions were the same as in our exploratory tests, with the exception that the duration of the runs at each load setting was increased to 7 hours and later to 14 hours.

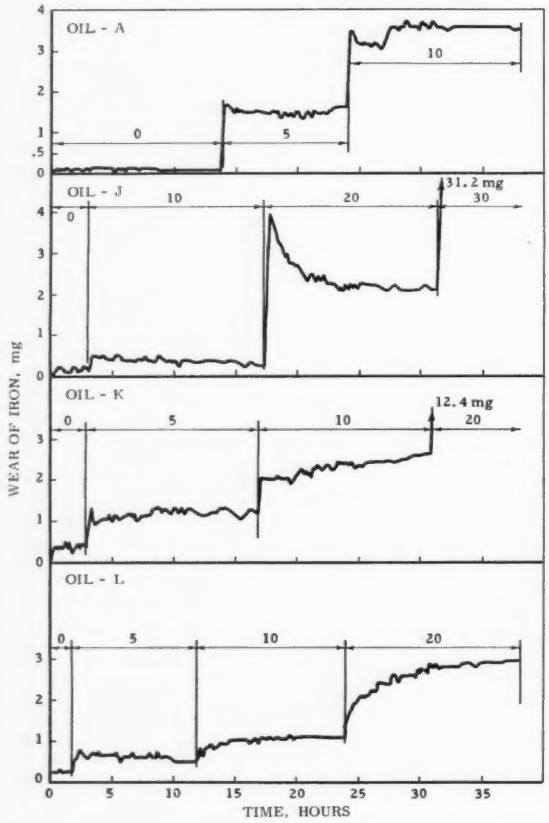

Fig. 6. Radioactive gear wear tests; mineral and E.P. compounded oils.

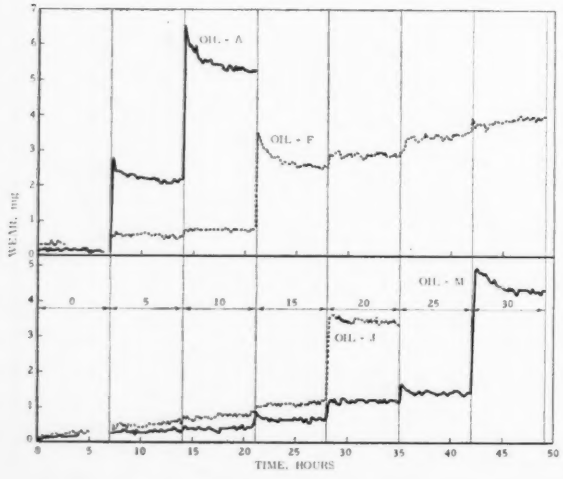

Fig. 7. Radioactive gear wear tests.

Wear Characteristics of Mineral, Synthetic and E.P. Compounded Oils. The oils investigated include a number of unreactive mineral oils, synthetic oils, and E.P. compounded oils. Their wear characteristics are presented in graph form in Figs. 5 to 7 and as radiographs in Figs. 8 & 9.

In general, the magnitude of wear caused by scoring was much higher than the wear caused by abrasion or chemical corrosion. Often, when scoring was severe, the counting rate increased instantaneously to a very high level and then gradually dropped to a lower value. This decrease in counting rate after scoring is due to the gradual distribution of the slug of worn particles throughout the

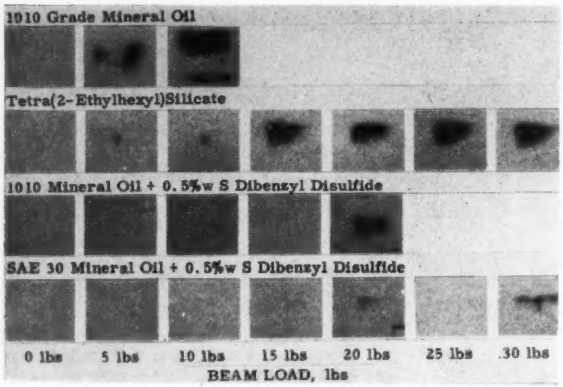

Fig. 8. Radiographs of teeth working surfaces.

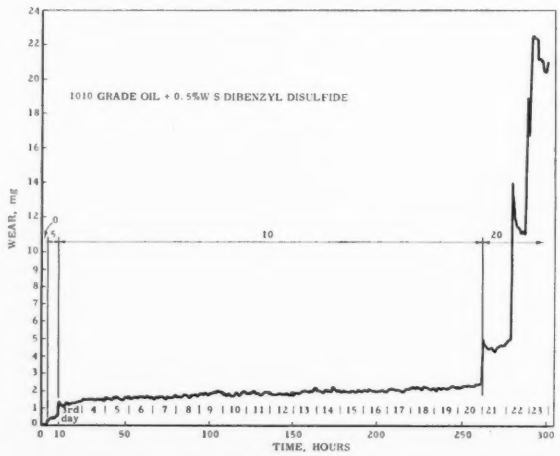

Fig. 9. Long-term radioactive gear-wear test.

total volume of the test oil, and by gradual settling of the larger particles. To minimize such settling, the circulating system was constructed in straight lines and pockets and sharp corners were avoided.

The wear behavior of a mineral oil and several synthetic oils is compared in Fig. 5. All of these oils are approximately of the same viscosity. The 1010 grade mineral oil (A) gave wear by scoring only, because wear occurred at the time when the load was first applied and was characterized by a sharp and instantaneous rise. Synthetic oils, such as water-soluble polyglycol (D), and diester di(2-ethylhexyl)sebacate (B), acted similarly to the mineral oil, although the magnitude of wear at comparable loads was somewhat smaller than with the mineral oil. With an oil-soluble polyglycol (C), a steady increase in wear observed at 5 pounds load was attributed to wear by corrosion, because microscopic examination revealed no sign of abrasion. Tri(2-ethylhexyl) phosphate (E) also showed a steady rise in wear at 10, 20, and 30 pounds load. The presence of abrasion marks suggests that wear with this oil was caused by abrasion. The wear chart for tetra(2-ethylhexyl) silicate (F) indicates a practical absence of wear up to 30 pounds load, where scoring occurred.

Wear characteristics of E.P. compounded oils are given in Fig. 6. The base oil of each lubricant was 1010 grade mineral oil (A), and the amount of compounding material was 0.5% by weight. With dibenzyl disulfide (J), wear was caused by scoring only. With chlorinated wax (K), besides light scoring at the beginning of the runs at each load, a steady increase in wear was noted. The absence of abrasion marks indicates that the cause of this wear was chemical corrosion. With tricresyl phosphate (L) the gradual increase in wear was attributed to abrasion, because microscopic examination revealed the presence of abrasion marks.

To provide the data for metal transfer studies, the tests with several oils were repeated. At the end of the run at each load the inert gear was removed from the machine, washed with a solvent, and radiographed in a light-proof cabinet with Eastman Kodak Industrial X-ray film, Type K. The exposure time was 16 hours. The wear occurring in these tests is presented in Fig. 7 and radiographs are given in Fig. 8. Comparison of the wear performance of these oils with the occurrence of metal transfer shows that metal transfer took place only at scoring. Light abrasion, which occurred in this test with 1010 grade oil compounded with 0.5%w S dibenzyl disulfide (J), was not accompanied by metal transfer. This in turn indicates that surfaces may be abraded without complete destruction of the lubricating film. It is of interest also that light localized scoring, as occurred for example with tetra-(2-ethylhexyl) silicate (F) at 5 pounds load, or with SAE 30 oil compounded with dibenzyl disulfide (M) at 20 pounds load, was detected by radiographs and wear data.

All of the above-described tests were of comparatively short duration, and gave no indication that time, as such, has any effect on wear of gears except through corrosion or abrasion. To determine the effect of time, several long-term tests were made. One of such tests, performed with 1010 grade oil compounded with 0.5%w S dibenzyl disulfide, is presented in Fig. 9. The tests were run for 250 hours at 10 pounds beam load, corresponding to 720 pounds per inch of face or a K-factor of 454. Yet the total wear during this period, which was of abrasive nature, was only 1.15 milligrams of iron.

Simple calculations show that this amount of worn iron corresponds to a layer on the working tooth surface about 10^{-6} inches thick. From a practical standpoint, wear of such magnitude can be considered insignificant, and the long-term tests thus show that accurately manufactured gears, when properly aligned and operated within their fatigue limits, can render long trouble-free service.

In general the employment of the radioactive tracer technique not only gave accurate and continuous measurements of wear, but also permitted us to define the mechanisms of wear by subdividing them into wear by scoring, wear by abrasion, and wear by chemical corrosion. Such a knowledge, in turn, adds to the understanding of the mechanism of gear lubrication and could be beneficial in the work of development of the gear lubricants.

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ACKNOWLEDGEMENT. The author wishes to express his appreciation to the Department of the Navy, Bureau of Aeronautics, under whose sponsorship this work has been conducted, for permission to publish this information.

COMMENTARY by P. H. Scheffler (Westinghouse Electric Corp.). Mr. Borsoff's findings, that gear wear is almost entirely due to scoring when lubricated with unreactive mineral oils and synthetic oils, agrees with results of tests made on Ryder Lubricant Test Gears at the Westinghouse Electric Corp., Aviation Gas Turbine Laboratory. These tests also showed that scoring occurs almost immediately, and that duration of operation has very little effect upon the degree or amount of scoring.

The radiographs of the inert gear teeth showing that metal transfer occurs with scoring, supports our belief that scoring is the result of welding of the contact surfaces. It can be shown analytically that when the oil film breaks down, the instantaneous temperature at the tooth contact point is high enough to cause melting of the surface. Briefly, the heat generated is the product of the normal tooth load, relative sliding velocity, and the coefficient of friction of the contacting surfaces. While the amount of heat is not large, it is concentrated over a small area which is a function of the surface finish and the Hertz deformation area. Further, it is distributed over this area proportionally to the Hertz stress distribution. Thus the maximum rate of heat flow per unit area is large enough to raise the temperature of the surface to its melting point.

From this it follows that steps to reduce the amount of scoring will include: (1) Tooth profile modification to reduce the maximum value of relative sliding velocity; (2) Control of tooth spacing error and profile variations to reduce the maximum tooth loadings; (3) Use of a lubricant with suitable film strength for the load to be carried to re-

duce the coefficient of friction; (4) Metallurgical control of the tooth surface to provide the highest melting point consistent with the required hardness.

AUTHOR'S CLOSURE. It is gratifying to note that the results of the tests performed at the Aviation Gas Turbine Laboratory, and those by the author, substantiate each other. Mr. Scheffler's explanation of the nature of scoring by melting of the top layers of metal is similar to that of the author.

The author agrees completely with Mr. Scheffler's suggestions for practical ways to reduce scoring. In fact, the tests for the effect of profile modification and the effect of various errors were performed at the Shell Development Laboratory, and the results are in full agreement with Mr. Scheffler's conclusions.

Book Reviews

Machine Design (2nd Ed.), by P. H. Black; McGraw-Hill Book Co., New York, 1955; 471 pages, over 400 illustrations, price $7.50. (Reviewed by A. R. Black, Shell Oil Co.)

This book fulfills its expressed purpose in providing a text book for the advanced student in machine design, and a practical reference book for the practicing designer.

It combines in one volume the related subjects of engineering design, materials, and methods of fabrication. Particularly gratifying to the lubrication engineer is the author's recognition of the lubricant as a material of construction, and his practical approach to sleeve bearing design. We could wish only for a somewhat broader coverage of lubricants, particularly the modern developments, to match the up-to-date character of the remainder of the volume.

Standard Methods For Testing Petroleum & Its Products (14 Ed.); The Institute of Petroleum, 26 Portland Place, London W1, England, 1955; 688 pages. (Reviewed by A. C. West, California Research Corp.)

This book is the British counterpart of the ASTM Methods Manual. Particular effort has been made to resolve minor differences between the IP and ASTM methods, and the ASTM method is always cross-referenced. The book should be of value to personnel closely associated with British marketing and petroleum processing.

IP Engine Test Methods For Rating Fuels; The Institute of Petroleum, London, 1955; 84 pages. (Reviewed by A. C. West.)

This is a small separate volume covering the engine test methods for rating motor, aviation, and Diesel fuels. The test procedures are cross-referenced to the technically identical ASTM procedure. The book should be of interest to personnel closely associated with British marketing; however, it contains little that is not covered in the ASTM counterpart.

Automatic Transmissions, by The Instructional Staff of the Commercial Trades Institute, Chicago; McGraw-Hill Book Co., New York, 1955; 406 pages. (Reviewed by A. C. West.)

The growing use of automatic transmissions in passenger cars, trucks, and busses, together with the ever-increasing interrelationships between transmission, engine, ignition, and even braking systems, makes it imperative for the auto mechanic to become familiar with the principles governing the operation of these mechanical devices. This book has three main objectives: (1) to describe the basic scientific principles of automatic transmission, (2) to present a simplified discussion of these principles, and (3) to illustrate the application of these principles in typical transmissions.

Following a brief historical review of the general problem of connecting an internal combustion engine to the rear wheels of a vehicle, several chapters are devoted to basic hydraulics and mechanics. These chapters are well-written to appeal to automotive servicemen. Planetary gear systems, hydraulics, and properties of fluids in motion are reviewed as background for the discussion which follows on fluid couplings and various types of torque converters.

The second half of the book describes in detail the construction, operation, adjustment, and trouble-shooting of the automatic transmissions on all of the popular makes of automobiles. Unfortunately, only transmissions up to 1952 are covered. The progress in automatic transmission design in the past four years makes this second half of the book somewhat out-of-date. For the details of specific automatic transmissions, the manufacturers' shop manuals provide more complete and, certainly, more up-to-date sources of information. In spite of the weakness of the book in this regard, the excellent discussion of the principles of operation of automatic transmissions covered in the early part of the book makes it a worthwhile edition for automotive mechanics, both professional and amateur.

Bearing Lubrication Analysis, by R. R. Slaymaker; John Wiley & Sons, New York, 1955; price $5.00. (Reviewed by B. H. Jennings, Northwestern University.)

This book, covering the general principles of lubrication analysis in bearing design, deals primarily with sleeve bearings which are the most common type found in machines. It is designed for the semi-advanced student, as well as for bearing designers.

Although it starts out with hydrodynamic theory as a basis for analysis, it gives a down-to-earth treatment of the various problems involved in bearing design, considering lubricants, bearing materials, clearances, proportioning, heat dissipation, and other design features.

This book should be a valuable addition to the library of anyone concerned with the problem of sleeve bearing design and utilization. The author has a clear and lucid style, and the diagrams and illustrations are easy to read.

A Radioactive Study Of The Metal-Cutting Process

by I. Finnie* & E. Rabinowicz**

Autoradiographs were taken of tools used to cut radioactive copper. The variation of metal transfer, to the tool from the copper, with rake angle, depth of cut, and cutting fluid was evaluated. It was found that the amount of metal transfer to the cutting face was of the same order of magnitude as that produced in a sliding experiment under analogous conditions. Examination of the distribution of copper fragments along the tool-chip interface showed that almost no fragments were found on the clearance face of the tool. The measured friction coefficient did not appear to have any direct relation to the other variables of the experiment.

When using radiotracer techniques to study the metal-cutting process, either the cutting tool, the lubricant, or the workpiece may be made radioactive, and the purpose of the investigation will determine the actual choice. Thus, using a radioactive cutting fluid, Loewen[1] showed that a cutting fluid could penetrate to the tip of the tool, while Shaw and Strang[2] used a radioactive workpiece to show that some metal transfer occurs between work and tool even when very effective cutting fluids are used.

For quantitative studies of tool wear the logical choice is a radioactive tool, and this method has been used by Merchant, Ernst, and Krabacher,[3] and by Colding and Erwell.[4] However, it was decided that, to study the friction and wear process along the chip-tool interface, a radioactive workpiece was preferable, since the transferred fragments would remain in a localized position on the tool rather than scattered among the chips. In addition, use of a large number of non-radioactive tools would permit greater flexibility in changing the variables of the tests.

Thus, we planned to carry out tests in which the tool angle, depth of cut, and nature of the lubricant were systematically varied, and to examine the tools for any radioactive material picked up on both the cutting and clearance faces. At the same time, the amount of metal transfer would be compared with that obtained in sliding experiments between a similar pair of metals.

EXPERIMENTAL PROCEDURE. (1) Mechanical. The conventional picture of two-dimensional continuous cutting is as shown in Fig. 1.

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(This paper was co-sponsored by the ASLE Technical Committees on Bearings & Bearing Lubrication, and Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 13, 1955.)

From the measured forces we can obtain an average coefficient of friction f for the tool-chip interface

$$f = \frac{F_v + F_h \tan \alpha}{F_h - F_v \tan \alpha} \quad (1)$$

it being assumed that no contribution to the cutting forces arises from possible rubbing along the clearance face. This has recently been queried,[5] and one of the purposes of the present investigation was to throw light on this point.

For the cutting tests a copper bar, length 3/8" and width 1/4", was cut by a clean sharp tool made of grade 18-4-1 high speed steel, and the cutting forces were measured by a strain gage dynamometer. After the cutting experiments, an autoradiograph was taken of the tool to study the transferred radioactivity.

(2) Radioactive. Copper was selected as being a very suitable metal for these studies. It machines with little trouble from built-up edge, gives a continuous chip, and its strong activity when irradiated enables autoradiographs to be of short exposure. Health precautions are fairly simple due to the short half-life, 12.8 hours, of its radioactive isotope Cu^{64}.

The slab of copper was irradiated in a nuclear pile at the Brookhaven National Laboratory. On being returned to M.I.T., the copper was cut in a two-dimensional planing operation, a single cut being made by each of a succession of clean, sharp tools. Autoradiographs were taken alternately of the two faces of each tool. The method of holding

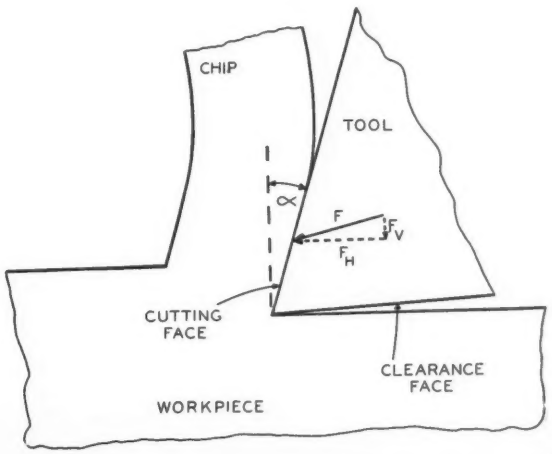

Fig. 1. The geometry of the cutting process.

the photographic film on the tool face is shown in Fig. 2.

Finally, for purposes of calibration (as will be discussed below) autoradiographs were taken of the workpiece itself, both bare and shielded by thin copper strip.

CALCULATION OF METAL TRANSFER. Once a correction has been made for the time factor introduced by the exponential decay of the radioactivity, the blackening of the film will be proportional to the thickness of the picked-up layer of copper. The product of film intensity and the area blackened will be proportional to the volume of metal transferred to the tool from the chip. To obtain a calibration relating the film blackening to the thickness of pick-up, use can be made of the autoradiographs of the copper block itself. As the block is thick, it will shield a considerable proportion of its own radiation, and its effective thickness for film blackening must be determined.

To a first approximation, we may regard the intensity of radiation in the block as falling off exponentially with distance, and the block as infinitely thick. Hence, the contributed intensity I at the surface, produced by a layer at a distance x inside the block, is

$$I = I_o e^{-x/x_o} dx \tag{2}$$

where I_o is the intensity produced by a layer at the surface, and x_o is a constant.

The total intensity on the film due to the radioactive block is

$$A_o = \int_o^\infty I_o e^{-x/x_o} dx = I_o x_o \tag{3}$$

With a shield of thickness a of the same material but non-radioactive, in front of the block the intensity is

$$A_a = \int_a^\infty I_o e^{-x/x_o} dx = I_o x_o e^{-x_a/x_o} \tag{4}$$

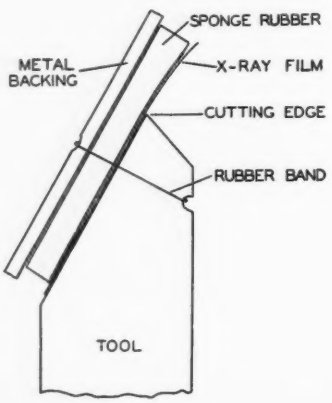

Fig. 2. Method of holding X-ray film during autoradiography.

and hence

$$x_o = x_a / ln(A_o/A_a) \tag{5}$$

By experiment it was found that with $x_o = 0.0063''$ the shielded autoradiograph had 1/5 the intensity of the unshielded; i.e., exposure of the unshielded copper block for time t produced the same amount of blackening as exposure of the shielded copper block for a time $5t$. Then $A_o/A_a = 5$ and $x_o = 0.0039''$ or 0.010 cm, and hence the infinite block with self-shielding gives the same intensity as would a sheet of thickness 0.01 cm not subject to shielding, and the other autoradiographs can be calibrated accordingly.

A series of exposures of the copper block was made, the time of exposure being t, $3t$, $10t$, $30t$, $100t$, and the tool autoradiographs were compared with them. In this way, by matching the exposure intensities and allowing for the time scale, an estimate could be made of the thickness of pick-up in each autoradiograph. Many of the autoradiographs have non-uniform blackening. The procedure, in that case, was to obtain by eye an average intensity over the area of exposure.

RESULTS. (A) Qualitative. In the thirty or so tools tested several tendencies are manifest.

1. Pick-up consists of a large number of particles spread out over the whole region of contact (Fig. 3).
2. A definite trend exists for pick-up to be most pronounced away from the cutting edge (Fig. 4).
3. If the tool surface is at all rough, pick-up

Fig. 3. Cutting face autoradiograph. Rake angle 25°; depth of cut .010"; no cutting fluid.

Fig. 4. Cutting face autoradiograph. Rake angle 45°; depth of cut .015"; with carbon tetrachloride.

Fig. 5. Cutting face autoradiograph. Rake angle 45°; depth of cut .015"; with carbon tetrachloride.

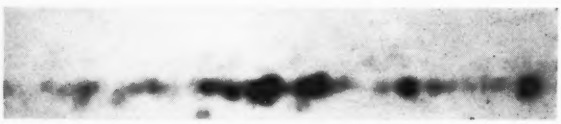

Fig. 6. Clearance face autoradiograph. Same tool as shown in Fig. 5, but radioactive exposure three times as great.

Table I Fluid	Metal Transfer	f
None	8×10^{-6}g	0.75
CCl_4	1.2	0.53
C_2H_5OH	0.2	0.67

Table II Rake Angle	Metal Transfer	f
25°	1.4×10^{-6}g	0.35
35°	1.2	0.53
45°	1.0	0.87

Table III Depth of Cut	Metal Transfer	
0.005"	0.5×10^{-6}g	
0.010"	1.2	
0.015"	2.0	

tends to concentrate at the tops of the most prominent ridges (Fig. 5).

4. Radioactivity along the clearance face is negligible compared to that on the tool-chip interface. A typical pair of autoradiographs is shown in Figs. 5 & 6. The amount of blackening on the clearance face is that expected due to the radiation from the cutting face. The majority of the particles in the clearance face autoradiograph can be recognized as particles also in the cutting face autoradiograph, and probably radiating from there.

(B) Quantitative. It is estimated that the metal transfer calculations are subject to an error of about 30%, and the coefficient of friction calculations about 10%.

1. Use of a cutting fluid. Depth of cut 0.010"; rake angle 35° (Table I). We note that while the alcohol produces a higher coefficient of friction than the carbon tetrachloride, the metal transfer is less.

2. Varying rake angle. Depth of cut 0.010"; fluid carbon tetrachloride (Table II). The wear results are very much the same, and the differences are not significant. The friction coefficients, however, vary widely.

3. Varying depth of cut. Fluid carbon tetrachloride; average pick-up for 25°, 35°, and 45° rake angles (Table III). The metal transfer is approximately proportional to the depth of cut.

Comparison with metal transfer rates obtained during sliding. A typical result for a test with no fluid, 35° rake angle, and 0.01" depth of cut may be compared with results reported by Rabinowicz and Tabor,[6] who found a metal transfer rate of 10^{-6} g/cm for copper sliding on mild steel, clean, under a load of 2 kg at slow speeds of sliding, the diameter of the circular contact area being 2.0×10^{-2} cm. In the cutting experiment, the metal transfer was 8.10^{-6} g, the contact area ¼" × 0.0074", and length of cut ⅜".

To compare the two rates of pick-up we may calculate the wear constant K which is the probability that two metal asperities in contact will produce a wear fragment (Archard[7]). The value for this constant may be written in the form

$$\frac{K}{3} = \frac{\text{Volume of material transferred}}{\text{Contact area} \times \text{sliding distance}} \qquad (6)$$

For the sliding experiment K is found to be 12×10^{-4}; for the cutting experiment $K = 3 \times 10^{-4}$. These values are of the same order of magnitude, and this suggests that the wear mechanism is essentially the same in both cases. The reason for the lower K value in the metal cutting case is that copper fragments on the tool tend to be removed as well as deposited while cutting continues, and thus the measured amount of transfer will tend to be too low. A comparable behavior does not exist in the sliding case, as the upper rider always travels over a different part of tool surface.

CONCLUSIONS. 1. There are systematic variations in sliding conditions along the tool chip contact region, and it is a crude approximation to regard conditions as uniform.

2. Very little action takes place along the clearance face when sharp tools are used. This is in agreement with the finding of Colding and Erwell,[4] who found very little activity transferred from a radioactive tool to a non-radioactive workpiece.

3. Pick-up varies linearly with the depth of cut and this is what we would expect, since the process is in each case the same but on a different scale.

4. Wear varies very little with rake angle while friction varies greatly, and this is an indication of the inadequacy of the concept of one coefficient of friction to characterize the whole sliding process in cutting.

5. Lubricants drastically reduce the metal transfer and, to a smaller extent, the coefficient of friction. It is to be noted that the lubricant which decreases wear the most will not necessarily have the greatest effect on the coefficient of friction, and this supports the conclusion of Rabinowicz and Tabor.[6]

6. The wear mechanism by which fragments are transferred from the chip to the tool is essentially that by which metal transfer takes place during sliding.

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COMMENTARY by A. O. Schmidt, B. F. Turkovich & J. R. Roubik (Kearney & Trecker Corporation). Although radioactive tracer technique has been used extensively in the study of wear and sliding phenomena, the authors' paper presents a new viewpoint. The use of a radioactive workpiece is certainly a convenient approach to investigating metal transfer from chip to tool in a machining operation. It remains to evaluate the significance of this particular kind of metal transfer in relation to tool wear and the overall metal-cutting picture.

The cutting speeds used in the experiments are of interest and have not been given. If the relative sliding velocity is sufficiently low, variation of that velocity, such as chip flow velocity over the tool face, should have no apparent influence on the rate of metal transfer. Since the authors state that a two-dimensional planing operation was used, and since they make a comparison between sliding results at low speeds and cutting results reported in this paper, it is safe to assume that the cutting speeds used were comparatively low.

At the clearance face an abrasion takes place which, in some cases, may be similar to a grinding action. The workpiece passes the clearance face at "cutting speed," which is about 3 times higher than the speed at which the chip slides over the tool. From other tests, we know that there is less adherence of workpiece material to the tool face when the cutting speed is higher. The same applies to the clearance face.

With an increase in sliding distance or total apparent length of contact between sliding surfaces, there will be an increase in the amount of metal transfer. The measured values of metal transfer reported in the paper represent the result of a complete cut and, hence, involve a sliding distance or length of formed chip passing over the tool surface. It is known that the chip length ratio can be expressed as a function of feed (depth of cut, in two-dimensional planing) and rake angle for a given tool-workpiece pair at constant cutting speed. A variation of 300 percent in feed (depth of cut) can be accompanied by almost as great a variation in chip-thickness ratio or, more appropriately, chip-length ratio. The reported data, then, apparently confirm the conclusion that the amount of metal transfer is a monotonic function of sliding length. If the contact area pressure were constant with variation of feed (depth of cut), the function would be linear.

That metal transfer is most pronounced away from the cutting edge seems most reasonable, when one considers the common phenomenon of cratering and the temperature distribution over the tool face (Trent, Proceedings of British Institute of Mechanical Engineers, Vol. 166, 1952). An increase in rake angle (more positive) is associated with higher chip thickness ratio, i.e., longer chips, and a reduction in cutting force. The combined effect may produce little variation insofar as the amount of metal transfer is concerned, but the variation in reported values has a reasonable trend.

Understanding of the influence and true role of a cutting fluid in metal-cutting is, as yet, clouded. According to the authors' test data, metal transfer and also coefficient of friction are influenced by the cutting fluid. Consideration of the following ratios of properties of ethyl alcohol to those of carbon tetrachloride: density, 1:2; latent head of vaporization, 5:1; and specific heat, 2.5:1, suggests the possibility that the heat transfer capacities may have exerted a predominant influence in the reported results. Presentation of the force measurement values taken would have enhanced the paper.

To further clarify some of the questions answered in, and questions arising from the paper, it would seem to be of value if similar radioactive tracer tests could be repeated, but with tools that had already done an amount of cutting (equivalent to ¼ or ½ a reasonable tool life), and sliders which had already done an equivalent amount of sliding with non-radioactive material. This suggestion is made since the amount of cutting which has been done in the reported tests is actually very small, which is in itself an advantage. In tests of the type suggested, the same small quantity of radioactive material as consumed in the reported tests would be used. There is a need for using newly-sharp tools when investigating the effect of tool geometry, but, on the other hand, most cutting is actually done with tools in a state of wear far removed from the newly-sharp condition. With such tests it should be possible to determine metal transfer to the flank of the tool after a wear land has been established, in which case a contribution to the cutting forces may arise from this source. It may be practical to shield the other face of the tool, when autoradiographs are made of the flank face, to obtain metal transfer values for this face alone. It also may be possible to determine what effect previous metal transfer may have had, and whether or not the cutting fluids would have the same effect as with newly-sharp tools.

The technique of Finnie and Rabinowicz presents us with a tool to evaluate quickly the effects of tool surface finishes. In machining nonferrous materials, the finish on the tool has a pronounced influence upon the finish and accuracy produced on the workpiece, e.g., aluminum and copper alloys will build up quickly on the tool if the surface finish on it is not of high quality, or if the grinding marks interfere with chip flow. Further possibilities of applying this procedure to technical problems would be a comparison of the influence of different directions of grinding marks, and the effect of microstructure of workpiece upon the amount of metal being transferred to tool face.

The authors are to be commended for this contribution to the literature which some day will provide us with a universal metal-cutting theory.

COMMENTARY by R. P. Hunnicutt (General Motors Corporation). An important point brought out in this work is the poor correlation between the coefficient of friction and metal transfer. This is in agreement with work of Goodzeit, Roach, and Hunnicutt, in which the rubbing characteristics of several metallic elements were rated on a basis of surface damage rather than frictional coefficient. In this work thirty-nine elemental metals were placed in rubbing contact with iron, and both their frictional coefficients and the resulting surface damage evaluated. The results indicated that little, if any, correlation could be found between the two values, and that a more realistic appraisal of the rubbing contact could be had on the basis of a surface damage examination.

AUTHORS' CLOSURE. The authors thank Dr. Schmidt and his colleagues for their informative discussion. As they surmised, the cutting speeds used in the experiments had the low values of 1 inch per minute which permitted comparison with slow speed sliding experiments.

The low cutting speed used in these experiments and, consequently, low tool temperatures make a comparison with the results of Trent on tool cratering at higher cutting speeds rather quesionable. It may be that the non-uniform metal transfer along the tool-chip interface is due to the temperature distribution, but a non-uniform stress state might also be responsible.

The comparison of the physical properties of the two cutting fluids is a valuable adjunct to the paper. Certainly in evaluating cutting fluids, the physical as well as the chemical properties must be considered, although with the slow cutting speeds used in the tests we would expect the importance of the physical properties to be less important.

The experiments of Mr. Hunnicutt and his colleagues differed greatly from that of the authors. It is interesting, however, that they reached the same conclusion as to the inadequacy of a friction coefficient to characterize the sliding process.

11th ASLE Annual Meeting & Exhibit, Pittsburgh, **April 4-5-6**

The Rheodynamic Squeeze-Film

by F. Osterle, A. Charnes & E. Saibel*

Considering grease as a Bingham plastic, the problem of the load supported by a squeeze-film where side flow is neglected is solved with grease as the lubricant. This load is found to depend on a single dimensionless parameter, and to exceed the load which would be supported by a Newtonian fluid with the same mobility by an amount which approaches a constant as the squeeze velocity is increased.

Nomenclature

x	=	coordinate in plane of film
y	=	coordinate normal to the plane of film
b	=	width of bearing
h	=	film thickness
V	=	squeeze velocity
τ	=	shear stress
τ_o	=	yield shear stress
m	=	mobility of lubricant
u	=	lubricant velocity in x direction
p	=	lubricant pressure
Q	=	lubricant flow in x direction
W	=	load capacity per unit bearing length
ϵ	=	dimensionless parameter defined in text
β	=	dimensionless velocity parameter, $Vb/m\tau_o h^2$
λ	=	dimensionless load parameter, $Wh/\tau_o b^2$

It has been shown experimentally[1] that grease-lubricated bearings can operate under thick film conditions, but with a behavior different from that of oil-lubricated bearings. This difference is undoubtedly due to the dissimilarity in the rheological properties of a grease and an oil. Oil, of course, very closely approximates a Newtonian fluid, while it has been known for some time that grease does not.

Recent work[2,3] has shown that to a first approximation, grease approximates a Bingham plastic. Both a Newtonian fluid and a Bingham plastic exhibit a linear relationship between shear stress and rate of shear, but with a Bingham plastic a certain yield stress must be exceeded before flow begins. This difference in rheological behavior results in the presence within the plastic film of a non-shearing or core zone which materially affects the development of pressure.

Lubrication with Newtonian fluids is referred to as hydrodynamic, and it has been suggested that lubrication with a Bingham plastic be called rheodynamic. Milne[4] made an important contribution to rheodynamic lubrication theory by investigating certain aspects of the slider-bearing problem. He found that in some cases the load capacity would be greater than that which would result if an equivalent Newtonian fluid (i.e., one with the same mobility as the plastic) were used as the lubricant.

In this paper rheodynamic lubrication theory is extended to the problem of the squeeze-film. Solutions to this problem for the hydrodynamic case are well known, having been investigated originally by Reynolds.[5] The problem is of importance in the evaluation of unsteady motion effects in the lubrication of gears and journal-bearings.[6]

THEORY. Consider the squeeze-film configuration shown in Fig. 1. As the elements approach each other the lubricant is squeezed out from between them and a pressure distribution is established in the film to maintain the viscous flow. The resulting load capacity can be quite appreciable.

If we apply the law of momentum conservation to the lubricant, and make the usual assumptions of lubrication theory, plus the special assumption regarding the plastic lubricant that is at all times capable of transmitting pressure, there results the following fundamental relationship

$$\frac{\partial p}{\partial x} = \frac{\partial \tau}{\partial y} \tag{1}$$

For the plastic lubricant the shear stress is related to the velocity gradient in the following way (see Fig. 2)

$$\frac{\partial u}{\partial y} = m(\tau \mp \tau_o), \quad |\tau| > \tau_o \tag{2}$$

$$\frac{\partial u}{\partial y} = 0, \quad |\tau| < \tau_o$$

where the minus (plus) sign applies when the shear stress is plus (minus). The reduction of Eq. 2 to the Newtonian fluid case is obvious, the mobility now being replaced by the reciprocal of the viscosity.

Combining Eq. 1 & 2 we have

$$\frac{\partial^2 u}{\partial^2 y} = m \frac{\partial p}{\partial x}, \quad |\tau| > \tau_o \tag{3a}$$

$$\frac{\partial u}{\partial y} = 0, \quad |\tau| < \tau_o \tag{3b}$$

To proceed further it is necessary to delineate the shearing and non-shearing (core) zones in the film. Consider the film cross-section at a station x where the pressure gradient is, for the sake of argument, negative. Then, by Eq. 1 a shear stress profile as shown in Fig. 3 must exist, since by symmetry there can be no shear stress at the mid-plane of the film ($y = 0$). If there

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(This paper was sponsored by the ASLE Technical Committee on Bearings & Bearing Lubrication, and presented at the ASLE 10th Annual Meeting, Chicago, April 15, 1955.)

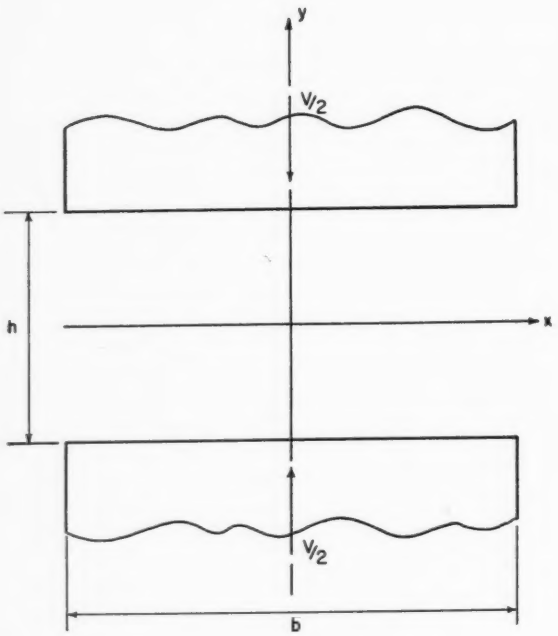

Fig. 1. The squeeze-film configuration.

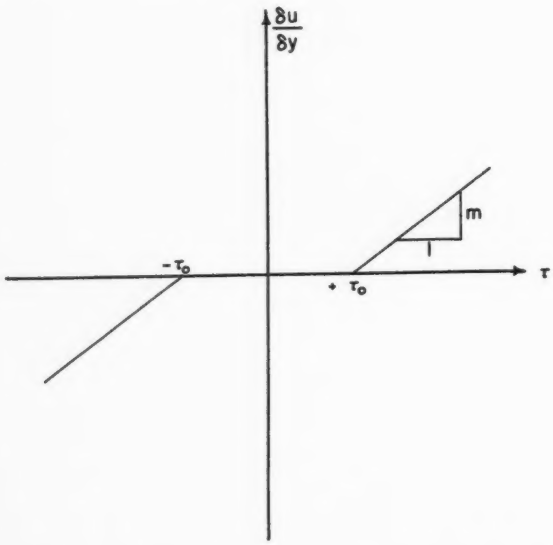

Fig. 2. Shear stress versus Strain rate for a Bingham plastic.

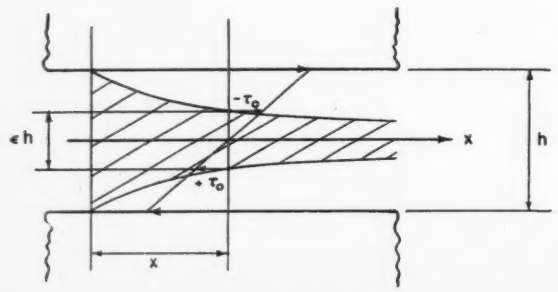

Fig. 3. Shear stress profile in squeeze-film.

is to be flow (i.e., $V \neq 0$), there must be shearing zones at all stations in the film other than $x = 0$. From the profile shown in Fig. 3, it is apparent that the shearing zones must be adjacent to the solid boundaries, and the core zone must be as shown cross-hatched in this figure. The shearing and core zones are separated by surfaces along which the shear stress is the yield value. We will define these surfaces by the equation

$$y = \pm \epsilon \frac{h}{2} \tag{4}$$

where the parameter ϵ is a function of x which must be determined.

In the shearing zones Eq. 3a governs the flow, while in the core zone Eq. 3b governs. These equations must be integrated for the velocity subject to the following conditions: (1) At the surfaces of the approaching elements the velocity is zero by the conventional no-slip hypothesis, (2) at the yield stress surfaces the velocity is continuous for shear stress finiteness, (3) at the yield stress surfaces the velocity gradient is continuous for shear stress compatibility. The lubricant flow defined by

$$Q \equiv \int_{-\frac{h}{2}}^{\frac{h}{2}} u \, dy \tag{5}$$

can then be determined by integrating the velocity expressions across the film. We obtain

$$Q = -\frac{mh^3}{24} \frac{dp}{dx} (1-\epsilon)^2 (2+\epsilon) \tag{6}$$

Applying the principle of conservation of mass to the lubricant between the approaching elements, there results the equation

$$\frac{dQ}{dx} = V \tag{7}$$

Integrating this once we obtain

$$Q = Vx \tag{8}$$

since by symmetry there is no flow at $x = 0$.

Combining Eqs. 6 & 8 we have

$$(1-\epsilon)^2 (2+\epsilon) \frac{dp}{dx} = -\frac{24Vx}{mh^3} \tag{9}$$

This is one equation for $p(x)$ in terms of the parameter ϵ. We need one more so that ϵ can be eliminated.

Applying the conservation of momentum law to the core zone in, say, the positive x region, there results by Eq. 1.

$$\frac{dp}{dx} = -\frac{\tau_o - (-\tau_o)}{\epsilon h}$$

or

$$\frac{dp}{dx} = -\frac{2\tau_o}{\epsilon h} \tag{10}$$

Now, Eqs. 9 & 10 completely describe the problem. Eliminating ϵ between these two equations, we have

$$\left(\frac{hp'}{\tau_o}\right) + 3 - \frac{4}{\left(\frac{hp'}{\tau_o}\right)^2} = -\frac{12\,Vx}{m\tau_o h^2} \tag{11}$$

where p' is the pressure gradient. Eq. 11 is then the fundamental equation governing the load capacity of the rheodynamic squeeze-film. It is readily solved for either very large or very small values of the characteristic velocity parameter β defined by

$$\beta \equiv \frac{Vb}{m\tau_o h^2} \tag{12}$$

These two limiting cases will be discussed first, then the case of intermediate values of β.

For very small values of β, Eq. 11 reduces to

$$\frac{hp_o'}{\tau_o} = -2 \tag{13}$$

where the subscript on p' refers to the smallness of β. Eq. 13 integrates to

$$p_o = \frac{\tau_o}{h}(b - 2|x|) \tag{14}$$

where the condition of zero pressure at the film edges has been utilized. If we define load capacity as

$$W \equiv 2\int_o^{\frac{b}{2}} p\,dx \tag{15}$$

then

$$W_o = \frac{b^2\tau_o}{2h} \tag{16}$$

For very large values of β, the non-linear term in Eq. 11 is negligibly small over most of the film. Neglecting it, Eq. 11 reduces to

$$\left(\frac{hp_\infty'}{\tau_o}\right) + 3 = -\frac{12\,Vx}{m\tau_o h^2} \tag{17}$$

where the subscript on p' refers to the largeness of β. Eq. 17 integrates to

$$p_\infty = \frac{3\tau_o}{2h}\left\{(b - 2|x|) + \frac{V}{m\tau_o h^2}(b^2 - 4x^2)\right\} \tag{18}$$

for the pressure and to

$$W_\infty = \frac{b^2\tau_o}{4h}\left[3 + \frac{4\,Vb}{m\tau_o h^2}\right] \tag{19}$$

for the load capacity.

Eq. 16 is the load capacity as β approaches zero, and Eq. 19 is the load capacity as β approaches infinity. If we let β approach zero in Eq. 19, we obtain a load capacity which is 50 percent high. As β increases, this error decreases.

It was discovered by comparison of Eq. 19 with a graphical solution of Eq. 11 that for a β greater than about 2, W_∞ is negligibly greater than W. Furthermore, examination of the parameter reveals that reasonable values of it will invariably be greater than 2. Thus for practical purposes, Eq. 19 is the solution to our problem.

If it should develop that accurate values of W are desired for β values less than 2, an iterative process can be used to solve Eq. 11 to any desired accuracy. The procedure is as follows: Determine p_∞ as above and substitute it into the non-linear term in Eq. 11. Solve the resulting linear equation for a corrected p_∞. Substitute this corrected p_∞ back into Eq. 11 to obtain a new corrected p_∞. Repeat this procedure as long as accuracy requires.

Archibald in private communication has pointed out that, if we recognize V as the negative derivative of h with respect to time, Eq. 19 gives the plate separation h as a function of time under conditions of constant load. He derives the following expression for the time it will take the plates to move from a separation of h_1 to a separation of h_2 under a constant load

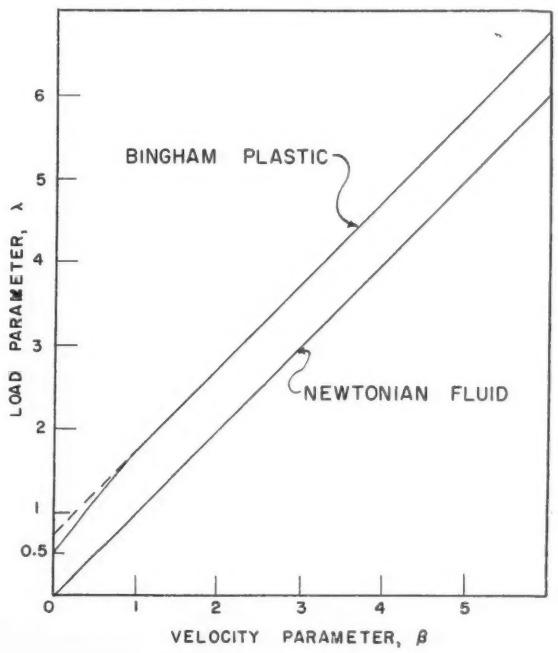

Fig. 4. Load parameter $Wh/\tau_o b^2$ versus Velocity parameter $Vb/m\tau_o h^2$

$$t_{12} = \frac{b}{9m\tau_o}\left[3\left(\frac{1}{h_1} - \frac{1}{h_2}\right) + \frac{8W_s}{b^2\tau_o} \ln\left(\frac{\frac{8W_s}{b^2\tau_o} - \frac{3}{h_1}}{\frac{8W_s}{b^2\tau_o} - \frac{3}{h_2}}\right)\right] \quad (20)$$

DISCUSSION. In Fig. 4 the dimensionless load capacity is plotted against β for the lubricant considered as a Bingham plastic. Also shown in this plot is the load capacity which would result if the lubricant were a Newtonian fluid with the same mobility. Since for a Newtonian fluid the load is proportional to the velocity, the yield shear stress contained in both the load and velocity parameters cancels out when Fig. 4 is interpreted for a Newtonian fluid. By comparing the two graphs, it is apparent that the effect of a non-zero yield stress is to raise the load capacity by an amount which approaches a constant as β increases. Thus, as β increases, the percentage increase in load capacity decreases. It is clearly evident from Fig. 4 that a Bingham plastic requires an initial load before flow begins, as contrasted with a Newtonian fluid which flows under any load however small.

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(Continued on p. 61)

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Exhibit Registration: The exhibit area is open to all registrants without charge, and is open to all visitors who register.

WEDNESDAY, APRIL 4, 1956

7:30 A.M. Breakfast for Authors and Chairmen
8:00 A.M. Registration
9:00 A.M. Official Opening of Exhibits
9:00 A.M. Annual Business Meeting

10:00-12:00 A.M.

SESSION 1A — URBAN ROOM — SPRAY APPLICATION OF LUBRICANTS

Session Chairman: J. D. Lykins, Wheeling Steel Corp.
Asst. Chairman: M. S. Hogue, U. S. Steel Corp.

The Spray and Dip Methods of Applying Lubricants to Wire Rope
D. B. Burroughs, Wire Rope Division, John A. Roebling & Sons Corp.

Spray Lubrication of Lubricants to Plain Roll Neck Bearings
C. M. Winn and J. S. Aarons, National Tube Division, U. S. Steel Corp.

SESSION 1B — MONONGAHELA ROOM — GENERAL INDUSTRIES SESSION I

Session Chairman: J. D. Hetchler, Archer-Daniels-Midland Co.
Asst. Chairman: H. B. Deakins, Gulf Oil Corp.

Chemical Machining and Grinding Fluids
A. W. Ackerman and F. E. Anderson, F. E. Anderson Oil Co., Portland, Conn.

2:00-5:00 P.M.

SESSION 2A — ALLEGHENY ROOM — STUDIES IN VISCOSITY I

Session Chairman: C. C. Currie, Dow Corning Corp., Midland, Mich.
Asst. Chairman: H. K. Siefers, United Oil Corp.

Surface Viscosity and Elasticity of Lubricating Oils
D. W. Criddle, California Research Corp.

Flow Analysis of Lubricants from Measurements with an Automatic Viscometer
R. N. Weltmann and P. W. Kuhns, NACA, Cleveland, Ohio

Problems in Hydrodynamic Lubrication
E. K. Gatcombe, U. S. Naval Post Graduate School, Monterey, Calif.

SESSION 2B — MONONGAHELA ROOM — PROPERTIES OF LUBRICANTS

Session Chairman: H. W. McCulloch, Jr., Shell Oil Co., Wood River, Ill.
Asst. Chairman: C. A. Bailey, U. S. Steel Corp.

Concentration Effects of Cutting Oil Additives in Performance Evaluation
A. Dorinson, Sinclair Research Laboratories.

The Dynamic Demulsibility Characteristics of Oils
E. W. Brennan and R. G. Moyer, Research and Development Laboratories, Pure Oil Co.

The Properties of Synthetic Ester Lubricants
M. Z. Fainman and R. S. Barnes, Standard Oil of Indiana

SESSION 2C — URBAN ROOM — INDUSTRIAL GEAR LUBRICATION

Session Chairman: T. W. Havely, Shell Oil Co.
Asst. Chairman: H. T. Anderson, Jones & Laughlin Steel Corp.

A Lubrication Engineer Surveys Steel Mill Gearing
A. E. Cichelli, Bethlehem Steel Co.

Design and Testing Considerations of Lubricants for Gear Applications
E. E. Shipley, General Electric Co.

Some Effects of Nuclear Radiation on Lubricants and Lubrication-Bearing and Gear Lubrication Tests
J. G. Carroll, R. O. Bolt, and G. H. Denison, California Research Corp.

SESSION 2D — FORT DUQUESNE ROOM — EDUCATION SESSION I

A. Fundamentals of Fluid-Film Lubrication
Session Chairman: D. D. Fuller, Mechanical Engineering Dept., Columbia U.

B. Fundamentals of Boundary Lubrication
Session Chairman: W. E. Campbell, Clevite Research Center, Clevite Corp.

THURSDAY, APRIL 5, 1956

7:30 A.M. Breakfast for Authors and Chairmen

9:00-12:00 A.M.

SESSION 3A — URBAN ROOM — STUDIES IN VISCOSITY II

Session Chairman: D. Godfrey, California Research Corp.
Asst. Chairman: C. E. Trautman, Gulf Research and Development Corp.

Grease Flow in Shielded Bearing Service
R. O'Halloran, J. J. Kolfenbach, and H. L. Leland, Esso Research and Engineering Co., Linden, N. J.

Capillary Viscometry of Lubricating Grease
L. C. Brunstrum and R. H. Leet, Standard Oil of Indiana

SESSION 3B—MONONGAHELA ROOM—EFFECT OF ENVIRONMENTAL FACTORS ON THE LUBRICANT

Session Chairman: A. W. Lindert, Standard Oil of Indiana

Asst. Chairman: J. I. C. Taylor, Sun Oil Co.

High Temperature Lubricant Studies
E. E. Klaus and M. R. Fenske, Petroleum Refining Laboratory, Pennsylvania State University

Effect of Engine Operation on Synthetic Gas Turbine Lubricants
T. F. Davidson and J. H. Way, Power Plant Laboratory, WADC, Wright-Patterson AFB, Ohio

Current Research in the Bacteriology of Soluble Oil Emulsions
H. Pivnick, L. R. Sabina, R. Samuel-Maharajah, and C. K. Fotopoulis, University of Nebraska

SESSION 3C — ALLEGHENY ROOM — RAILROAD JOURNAL BEARINGS

Session Chairman: L. E. Hoyer, American Brake Shoe Co .

Asst. Chairman: J. L. Duchene, Baldwin-Lima-Hamilton Corp.

Analysis of Factors Influencing Railroad Bearing Operating Temperatures under Service Conditions
W. M. Keller, Assoc. of American Railroads, and G. L. Pigman, Armour Research Foundation

Experimental Investigation on Railroad Journal Bearing Operation Characteristics
A. L. Busby and G. L. Pigman, Armour Research Foundation

Investigation of Oil Additives for Boundary Lubrication of Railroad Journal Bearings
A. Miller and A. A. Anderson, Armour Research Foundation

SESSION 3D — FORT DUQUESNE ROOM — EDUCATION SESSION II — LUBRICANTS

Session Chairman: C. A. Bailey, National Tube Div., U. S. Steel Co.

2:00-5:00 P.M.

SESSION 4A — ALLEGHENY ROOM — HYDRODYNAMIC BEARINGS

Session Chairman: E. R. Booser, General Electric Co.

Asst. Chairman: J. F. Osterle, Carnegie Institute of Technology

A Solution for the Finite Journal Bearing and Its Application to Analysis and Design
A. A. Raimondi and J. Boyd, Westinghouse Research Laboratories

Numerical Solution of Reynolds Equation for Sector Thrust Bearings
B. Sternlicht, General Electric Co., and H. J. Sneck, Mechanical Engineering Dept., Rensselaer Polytechnic Institute

Roll Neck Lubrication
J. C. Dixion, Cities Service Research and Development Co.

SESSION 4B — URBAN ROOM — PANEL: FIRE RESISTANT FLUIDS TODAY

Session Chairman: G. B. Miller, Racine Machinery & Hydraulics Inc.

Asst. Chairman: M. Parker, E. F. Houghton Co.

Phosphate Base Fluids
F. H. Langenfeld, Monsanto Chemical Co.

Aqueous Base Fluids
W. H. Millett, Carbide & Carbon Chemical Co.

Emulsions
R. G. Larson, Shell Oil Co.

(Additional panel members have been selected but not verified)

SESSION 4C — MONONGAHELA ROOM — METALWORKING LUBRICANTS

Session Chairman: E. J. Krabacher, Cincinnati Milling Machine Co.

Asst. Chairman: L. B. Sargent, Aluminum Company of America

A Discussion of the Heat Treater's Problem in Quenching Medium Selection Oils
C. R. Weir, Commonwealth Industries, Inc.

The Influence of Fluids on the Temperatures Obtained in Grinding
M. C. Shaw, Massachusetts Institute of Technology

Rolling Oils as Applied to Ferrous and Non-Ferrous Metals
S. A. Paradee, Archer-Daniels-Midland Co.

A Tapping Test for Evaluating Cutting Fluids
C. D. Flemming and L. H. Sudholz, Socony-Mobil Oil Co., Inc.

SESSION 4D — FORT DUQUESNE ROOM — EDUCATIONAL SESSION III — APPLICATION OF LUBRICANTS

Session Chairman: R. J. Torrens, Eastman Kodak Co.

FRIDAY, APRIL 6, 1956

7:30 A.M. Breakfast for the Authors and Chairmen

9:00-12:00 A.M.

SESSION 5A — ALLEGHENY ROOM — ROLLING CONTACT BEARINGS

Session Chairman: E. E. Wagner, Hoover Ball & Bearing Co.

Asst. Chairman: H. N. Kaufman, Westinghouse Research Laboratories

New Retainer Materials for Aircraft Gas Turbine Bearings
F. C. Wagner and J. T. Burwell, Horizons, Inc.

A Study of a Combustion Resistant Hydraulic Fluid
H. V. Cordiano, Brooklyn Navy Yards

SESSION 5B — MONONGAHELA ROOM — DISPOSAL OF WASTE LUBRICANTS AND COOLANTS WITH REFERENCE TO STREAM POLLUTION

Session Chairman: A. A. Paul, Brooks Oil Co.

Asst. Chairman: O. G. McCandless, American Rolling Mills Co.

The Disposal of Soluble Oil Waste
E. E. McClung, Van Straaten Chemical Co., Chicago, Ill.

Treatment of Oil Wastes from Machining Operation
M. F. Madarasz, Ford Motor Co.

SESSION 5C — URBAN ROOM — BOUNDARY LUBRICATION

Session Chairman: A. Dobry, Westinghouse Research Laboratories

Asst. Chairman: A. J. DeArdo, Aluminum Company of America

Surface Film Thickness and Boundary Lubrication
I. Ming Feng, Bendix Aviation Co.

A Transition Temperature for Surface Damage in Sliding Metallic Contact
L. F. Coffin, Jr., General Electric Co.

Boundary Lubrication Studies of Typical Fluoroesters
R. C. Bowers, R. L. Cottington, T. M. Thomas, and W. A. Zisman, Naval Research Laboratories

The Equilibrium Distribution of Transfer Fragments
E. Rabinowicz, Massachusetts Institute of Technology

SESSION 5D — FORT DUQUESNE ROOM — EDUCATIONAL SESSION IV — ROLLING CONTACT BEARINGS, SEALS, AND GEARS

Session Chairman: W. T. Everitt, Eastman Kodak Co.

2:00-5:00 P.M.

SESSION 6A — URBAN ROOM — SURFACE PHENOMENA IN LUBRICATION
Session Chairman: E. M. Kipp, Aluminum Company of America
Asst. Chairman: P. R. Eklund, Westinghouse Atomic Power Div.
Mechanism of "Free" Rolling Friction
D. Tabor, Cambridge University, United Kingdom (Presently, International Fellow, Stanford Research Institute)
Electron Diffraction in Lubrication Research
D. Godfrey, California Research Corp.
On a Criterion of Cutting Velocity in Metal Machining
F. F. Ling and E. Saibel, Carnegie Institute of Technology

SESSION 6B — MONONGAHELA ROOM — MAINTENANCE
Session Chairman: L. O. Witzenburg, Farvel Corp.
Asst. Chairman: W. G. Ritter, Ritter Engineering Co.
Crane Maintenance Program
T. E. Hughes, Morgan Engineering Co.
Movie: Taconite Processing (30 Min.) — Reserve Mining Co.
Lubrication Problems Connected with Taconite Operations.
C. R. Lewis, Republic Steel Co.

SESSION 6D — FORT DUQUESNE ROOM — EDUCATION SESSION V — PLANT LUBRICATION
Session Chairman: K. S. Smiley, E. I. du Pont de Nemours & Co.

LADIES' PROGRAM

Chairman: Mrs. J. Boyd

Wed. April 4	Registration	**9:00-12:00 PM**
	Luncheon, Carlton House	**Noon**
	Pittsburgh Railways Bus Tour of Pittsburgh	**1:30-4:30 PM**
Thurs. April 5	Tour of H. J. Heinz Company	**9:30-11:30 AM**
	Luncheon, Greater Pittsburgh Airport	**Noon**
	Tour of Greater Pittsburgh Airport	**1:30-3:00 PM**
Fri. April 6	Tour of Nationality Room—University of Pittsburgh and Phipps Conservatory (Oakland)	**Noon-2:15 PM**

LUBRICATION ENGINEERING COURSE

All sessions to be held in the Ft. Duquesne Room

Wed. April 4	SESSION 2DA—Fundamentals of Fluid-Film Lubrication	**2:00-5:00 PM**
	SESSION 2DB—Fundamentals of Boundary Lubrication	
Thurs. April 5	SESSION 3D—Lubricants	**9:00-12:00 AM**
Fri. April 6	SESSION 4D—Application of Lubricants	**2:00-5:00 PM**
	SESSION 5D—Rolling Contact Bearings, Seals, and Gears	**9:00-12:00 AM**
	SESSION 6D—Plant Lubrication	**2:00-5:00 PM**

Fee Schedule shown inside

Chairman: Professor B. G. Rightmire
Massachusetts Institute of Technology

A Study Of Fretting Wear In Mineral Oil

by D. Godfrey*

Experiments were conducted to determine the important factors of fretting in the presence of mineral oil. The results obtained with a hard steel ball vibrating in contact with a soft steel flat showed that, in the first few cycles, metal is transferred to the ball to cause galling and the softer metal is scored and plowed. The damage increases with number of cycles. Wear fragments, which are generated at a constant rate, accumulate in the oil surrounding the contact area and cause the mixture to become a viscous paste that is physically displaced from the contact area. From this point on, fretting is "unlubricated;" and brown films and cocoa-colored debris appear.

In unlubricated fretting, the rate of wear depends upon the relative humidity of the surrounding air. Maximum wear occurs at approximately 30% relative humidity. The physical properties of the debris change with humidity and apparently determine the rate of wear. In an investigation to evaluate lubricants for fretting, it is important to keep out of the "unlubricated" state and to hold humidity constant. Runs with a few representative additives show that certain additives do affect the area of the fretted spot and the occurrence of surface films.

The phenomenon of fretting has been established as a type of wear,[1, 2] which is caused by the slip or reciprocating sliding of low amplitude that occurs between surfaces of vibrating mechanisms. Since it is a type of wear, it should be prevented or mitigated by lubrication. The effectiveness of oils and greases in reducing the intensity of fretting has been summarized by Herbeck and Stroheker,[3] who concluded that "there is great conflict in the effect of specific lubricants." The purpose of this research was to determine some of the important factors involved in lubricated fretting by conducting fretting experiments in the presence of a mineral oil under carefully controlled conditions.

APPARATUS, MATERIALS & PROCEDURE. Experiments were conducted on an apparatus with constant amplitude, load, frequency, and temperature, with standard specimens (a hard steel ball and soft steel flat), and with pure mineral oil and mineral oil plus additives.

The apparatus (Fig. 1a) was designed to provide constant amplitude even though the friction force varied during fretting. The apparatus causes a one-half inch diameter ball to vibrate in contact with a flat (0.059 x 1 x 3 inches) with an amplitude of 0.004 inch, a load of 100 grams, and a frequency of 30 cycles per second. The rotary motion of the shaft (driven by an 1800-rpm synchronous motor) is changed to reciprocating motion in the ball socket base by means of an eccentric bearing and guide ring arrangement at the top of the shaft. The load that the ball exerts upward against the stationary flat is applied by a spring under the hinged ball socket. The whole apparatus is enclosed to permit control of the atmosphere surrounding the specimens.

The action that occurs between two steel specimens was viewed by means of the optical arrangement shown in Fig. 1b. Parallel light is directed toward the contact area and then into a contour projection lens. The lens magnifies and projects the side-view shadow of the contacting specimens onto a ground glass screen. At 60X magnification, the ball appears as a segment of a circle. The flat specimen surface is not evident, except that it reflects the ball shadow, giving the appearance of two contacting spheres. Fig. 1c, however, shows the shadow as a contacting plane and sphere. Any liquid placed on the specimens is held around the contact area by surface tension and assumes the shape of a plano-concave lens. The initial volume of lubricant was calculated from the measured diameter of the lubricant shadow and the known diameter of the ball.

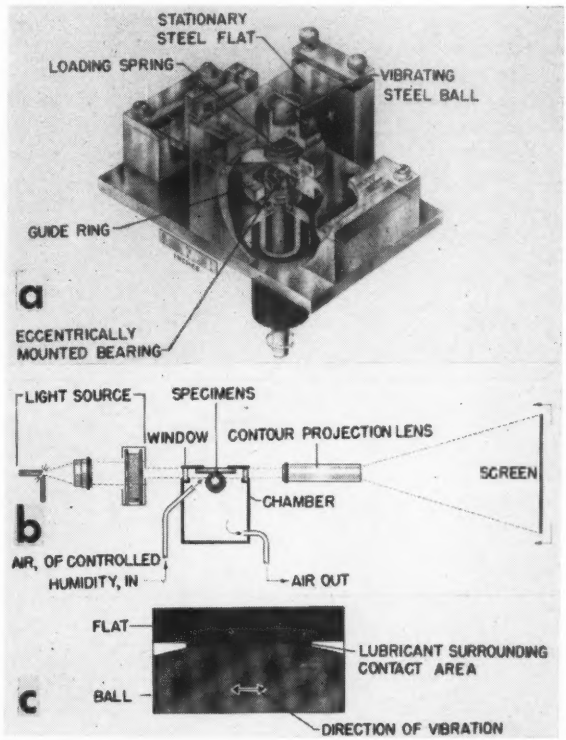

Fig. 1. Sketch of (a) fretting apparatus, (b) optical arrangement for viewing shadow of contact area, and (c) shadow of contact area.

*California Research Corp., P. O. Box 1627, Richmond 1, Calif. (Editor's Note: Mr. Godfrey conducted this research while in the employ of the National Advisory Committee for Aeronautics, Cleveland, Ohio.)

(This paper was co-sponsored by the ASLE Technical Committees on Bearings & Bearing Lubrication and Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 15, 1955.)

Without a lubricant, the fretting between the specimens was evident on the projected shadow from debris "dancing" out of the contact area.

The specimen material and preparation, the cleaning, and the mounting procedures were held constant throughout all runs. The one-half inch diameter balls were carbon steel case hardened to Rockwell C-59, obtained commercially as hardware type, Grade "A" balls. The one- by three-inch flats were cut from 0.07% carbon 16-gage cold-rolled sheet with a hardness of Rockwell B-55, and then ground. The ball and the flat were lightly abraded with 2/0 emery paper to give a uniform surface finish of 10-20 microinches and to abrade off any foreign material.

The abraded specimens were cleaned to make them grease-free (as evidenced by passing the water wet test) by the following procedure: scrubbed in fresh naphtha; wiped with a clean cloth; rinsed in freshly distilled benzene and then in 200-proof redistilled alcohol (both in Soxhlet extractor); dried with an air blower; cleaned anodically in a caustic bath; rinsed in redistilled water and then in redistilled 200-proof alcohol; and finally dried with an air blower. The specimens were immediately mounted in the apparatus, the ball socket of which was also cleaned for every run. This procedure is reported in detail in Appendix A of Ref. 4.

In order to control the volume of mineral oil surrounding the contact area, the mineral oil was diluted with benzene. One drop of a solution of 3.5% mineral oil in benzene applied to the ball would give approximately 0.001 ml of mineral oil surrounding the contact area. The mineral oil of USP grade was purified of polar compounds by percolating it through a series of columns containing layers of silica gel and fuller's earth. The resultant purified mineral oil had a viscosity of 28.0 centistokes at 100 F., and 4.7 centistokes at 210 F. Polar compounds were also removed from the stock benzene by the same percolating procedure. The additives were used as obtained commercially. All glass vessels, pipettes, etc., were cleaned by the usual chromic acid cleaning procedure.

The humidity of the air in the chamber housing the fretting apparatus was controlled. Commercial, activated alumina-type equipment dried the air to a dewpoint of at least —50 F. The outlet line was divided and part of the air was saturated by bubbling it (fritted inlet) through 30 cm of distilled water; the other part was run directly to a mixing chamber connected to the fretting apparatus. The humidity was controlled by adjustments of fine control valves in both the dry line and the saturated line. The humidity was read on a Serdex hygrometer enclosed and protected by a Lucite box with inlet, outlet, and dry-bulb thermometers inserted. The temperature of the air was held at 75±2 F. The hygrometer was checked frequently by a dewpoint potentiometer and found accurate within ±1.0% between 10% and 90% relative humidity. A dewpoint apparatus was used for relative humidities approaching 0% and 100%.

The standardized procedure, therefore, was as follows: adjusted humidity of air to desired value and allowed one-half hour for stabilization; cleaned specimens and mounted ball in apparatus; deposited the lubricant on the ball and let drain for 30 seconds; placed flat specimen in position; placed chamber over apparatus, established air flow and allowed five minutes for stabilization and benzene evaporation; measured diameter of lubricant on projected shadow; started ball vibrating and timed the number of cycles. At the end of the run, the specimens were separated and examined microscopically noting damage, and nature of lubricant, debris, or wear

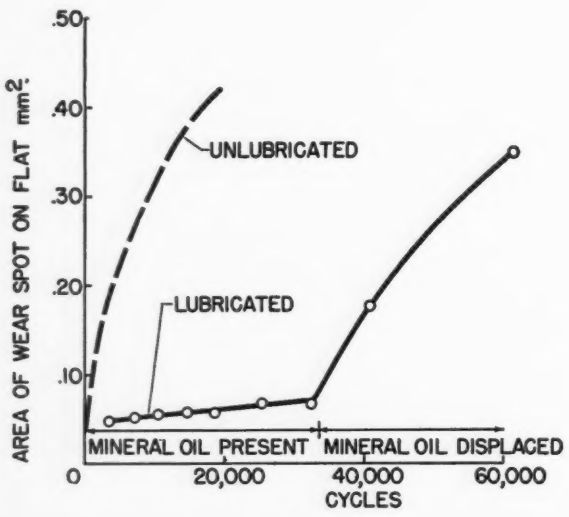

Fig. 2. Effect of cycles on fretting with constant volume of mineral oil surrounding the contact area; 30 percent relative humidity at 75 F.

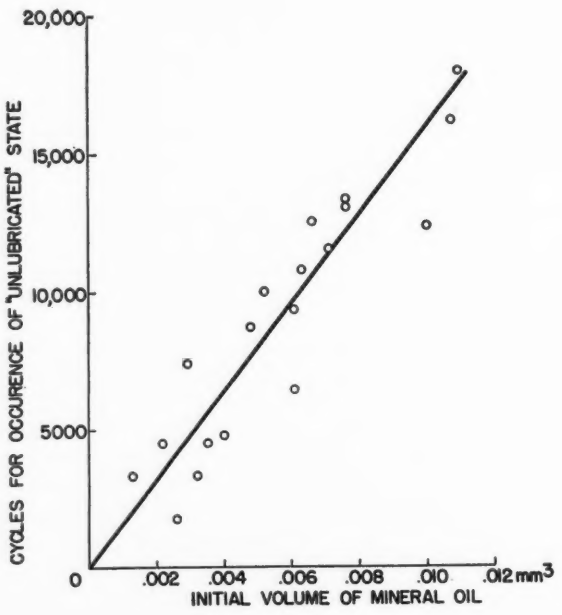

Fig. 3. Effect of initial volume of mineral oil surrounding contact area on number of cycles to occurrence of "unlubricated" state; 30 percent relative humidity at 75 F.

fragments. The diameter of the wear spot on the ball and the length of the major and minor axes of the ellipse-shaped wear spot on the flat were measured with a calibrated filar eyepiece in the microscope.

The area of the wear spot on the flat was used as a criterion of amount of wear or damage. The area was calculated from $\pi ab/4$, where a and b are the major and minor axes of the ellipse. In most cases, three identical runs were made with new specimens for each run. Every data point obtained is presented in the results so that the amount of scatter is shown.

RESULTS & DISCUSSION. (a) LUBRICATED. Mineral Oil. In the presence of mineral oil, the first half cycle of vibration damages both metal surfaces. The flat is scored and pick-up is evident on the ball. With continued vibration, wear fragments are generated (some are large and metallic and some are small, dark, and composed of αFe), which accumulate in the oil. The hard ball shows galling, and the soft flat shows a shallow concavity with parallel furrows. The steady accumulation of wear fragments gives the lubricant a "dirty oil" appearance after a few thousand cycles. Further accumulation of wear fragments causes the mixture to become a viscous paste. When the paste is sufficiently stiff to be displaced from the contact area, brown films and cocoa-colored debris appear that are characteristic of unlubricated fretting. From this point of "unlubricated" fretting when there is no lubricant in the real contact area, the rate of wear depends upon the relative humidity of the surrounding air, as will be shown later.

This effect of mineral oil on fretting is shown quantitatively in Fig. 2. The data reveal that, in the presence of a constant initial volume of mineral oil, wear increases uniformly (at a low wear rate) to the "unlubricated" state, whereupon the amount of wear suddenly increases and brown films and cocoa-colored debris appear.

The occurrence of this "unlubricated" condition depends upon the amount of mineral oil surrounding the contact area at the beginning of the run. The effect of initial volume of mineral oil on the number of cycles to "unlubricated" fretting is shown in Fig. 3.

Runs were made with relatively large quantities of mineral oil and relatively small number of cycles, so that the "unlubricated" state was not attained. The effect of humidity on the rate of fretting wear in the presence of mineral oil was determined, and the results are shown in Fig. 4. The data reveal that higher air humidities only slightly decreased the area of the wear spot on the flat. Microscopic examination of the wear spot after each run often revealed films on the wear surfaces at the higher humidities.

The specimen arrangement of these experiments represents local areas of contact between surfaces in vibrating machines. In the practical case, the paste of wear fragments and mineral oil that is formed around the local contact areas dams off the area from the remainder of the lubricant in the area and permits the "unlubricated" or locally dry state to occur early. Forced circulation of lubricant through the susceptible areas should be beneficial if the circulating lubricant removes wear fragments and thus prevents the paste formation.

Mineral Oil Plus Additives. A few representative additives were mixed with the mineral oil to determine their effect on fretting. The results are shown in Table I. The area of the wear spot on the flat is used as a measure of fretting damage. The additive octadecanol, which lubricates by physical adsorption,[5] had very little effect in reducing fretting wear. Stearic acid, which lubricates by chemisorption,[5] was more influential than octadecanol. However, 5% of sperm oil, which is a fatty acid derivative, was essentially ineffective. The results with the sulfur-containing additives, benzyl sulfide, and phenyl sulfide, showed that greater chemical reactivity was desirable in the case of additives which lubricate by sulfide films from chemical reactions with the metal surface. The 15% solution of benzyl sulfide caused more wear than the 5%. Examination of the lubricant after this 15% run revealed a quantity of black powder, apparently a reactive product caused by the high percentage

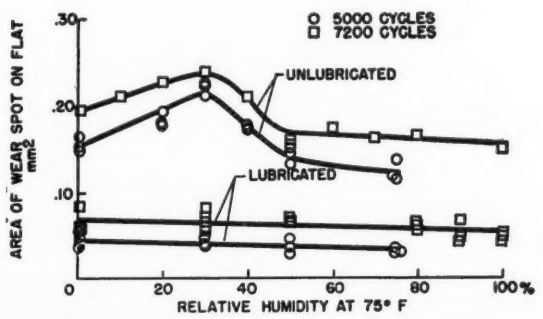

Fig. 4. Effect of relative humidity on area of wear spot on flat for unlubricated and lubricated fretting.

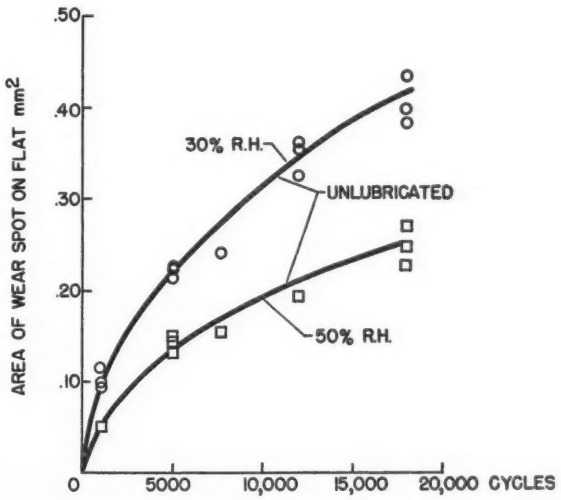

Fig. 5. Effect of percent relative humidity on unlubricated fretting.

of sulfur. The chlorine-containing additives, benzyl chloride, and the less reactive p-dichlorobenzene had little, if any, effect in reducing fretting wear; the surfaces were galled, and no reaction films were visible.

Phosphorus-containing additives mostly reduced fretting wear by generating reaction films on the surface. Extremely high percentages of tricresyl phosphate increased the wear; whereas, extremely high percentages of diethyl hydrogen phosphate decreased the wear. Five per cent of triethyl phosphorothioate appeared beneficial.

An extreme pressure commercial hypoid gear lubricant of unknown composition, run for comparison, reduced fretting wear only slightly.

These results indicate that certain additives do have an effect on area of wear spot and occurrence of surface films during fretting.

(b) UNLUBRICATED. Inasmuch as an "unlubricated" condition occurs with fretting in mineral oil, experiments were conducted with initially unlubricated specimens.

General Observations. Under the conditions of these unlubricated experiments, fretting starts with the very first vibration, as evidenced by the immediate appearance of loose debris in the projected shadow. The fretting forms a spot on the hard ball and a shadow concavity on the soft flat approximately the shape of an ellipse. Incrusted debris and films are found on both surfaces. Loose dry powder or debris is formed and extruded from the contact area.

Effect of Humidity of Surrounding Air. Variation of relative humidity from a value approaching zero to a value near saturation has a marked effect on unlubricated fretting. The upper two curves of Fig. 4 show that, in 5000 or in 7200 cycles, the area of the wear spot on the flat passes through a maximum near 30% relative humidity. This effect would be accentuated if weight loss or volume of metal removed from the ball were used as damage criterion. Microscopic examination of the fretted areas after each run showed that the character of the surface of the metal, the films of compacted debris in the contact area (stain), and the loose debris were different at various humidities. In particular, at relative humidities of 10% or lower, the metal surface was irregularly rough, matte gray, and no stain was visible; the debris was coarse and dry appearing and colored purplish red. At relative humidities of 20%, 30%, and 40%, metallic bright furrows were evident on the surface and the surface was partly incrusted with debris which appeared as brown film; the loose debris was medium coarse, cocoa-colored, and was generated in voluminous amounts. At relative humidities of 50% or greater, the surfaces showed furrows and brown films which became mahogany or black with increasing humidity. The debris remained cocoa-colored at these higher humidities but was of finer texture and smaller quantity than debris obtained at lower humidities; the debris readily packed into close clearances.

The difference in the fretting wear rate is apparently a result of variation in physical characteristics of the fretting products with humidity. Wright[2] reported volume of metal removed during fretting at only three humidities, as follows: 17×10^{-6} cc at 0%; 2.5×10^{-6} cc at 45%; 6.5×10^{-6} cc at 100%. Uhlig et al[6] concluded that increasing humidity decreased specimen loss. An examination of Wright's data reveals that he may not have taken points at sufficiently close intervals to reveal the maximum wear near 30%. Uhlig's data points did not show any significant increase in specimen weight loss below about 40% relative humidity.

Table I. Effect of various additives in mineral oil on area of wear spot on flat specimen; 30 percent relative humidity at 75 F.; 7200 cycles.

Additive		Description of fretted spot	
Name	Per-cent	Average area on flat mm^2	Microscopic appearance
(None, unlubricated)	--	0.240	Furrows
(None, pure mineral oil)	0	.068	Furrows; galled, bright metallic
Octadecanol	5	0.051	Furrows; galled, bright metallic
Sperm Oil	5	0.054	Furrows; galled, bright metallic
Stearic acid	5	0.040	Smooth, film present
Phenyl sulfide	5	0.049	Smooth, film present
Benzyl sulfide	5	0.059	Smooth, film present
	15	0.099	Smooth, black powder, film present
p-dichloro-benzene	5	0.054	Furrows; galled, bright metallic
Benzyl chloride	5	0.061	Furrows; galled, bright metallic
	15	0.045	"
	25	0.057	"
Tricresyl phosphate	5	0.041	Smooth, film present
	25	0.086	Furrows, film present
Diethyl hydrogen phosphite	5	0.051	Furrows, film present
	25	0.037	Smooth, film present
	50	0.028	"
Triethyl phosphoro thioate	5	0.037	Smooth, film present
	15	0.049	"
Commercial E.P. lubricant		0.049	Smooth, film present

Uhlig et al[6] concluded that increasing humidity decreased specimen loss. An examination of Wright's data reveals that he may not have taken points at sufficiently close intervals to reveal the maximum wear near 30%. Uhlig's data points did not show any significant increase in specimen weight loss below about 40% relative humidity.

The effect of number of cycles on area of wear spot for relative humidities of 30% and 50% is shown in Fig. 5. These data show that at 30% the area of the wear spot is almost twice as great as it is at 50%. The curve for 30% parallels that part of the curve in Fig. 2 where the mineral oil has been displaced from the contact area. This parallelism shows that no lubricant was present in the contact area; therefore, the rate of fretting depends upon the humidity of the surrounding air.

CONCLUSIONS. During lubricated fretting in mineral oil, metal-metal contact is made and wear fragments of αFe are formed. However, the low-amplitude reciprocal nature of the action creates the special conditions described which lead to accelerated unlubricated wear. Investigations of fretting inhibition with liquid lubricants should,

therefore, be directed toward reduction of rate of initial wear fragment formation.

In an investigation of the effects of lubricants on fretting, several factors are important:

1. The rate at which wear fragments are generated (or metal removed) in the period of the test before brown films and cocoa-colored debris appear is the true evaluation of lubricant effectiveness.

2. Humidity of the air has a marked effect in unlubricated fretting.

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(1) W. E. Campbell: The Current Status of Fretting Corrosion. Special Pub. No. 144, ASTM Symposium on Fretting Corrosion, June 1952, pp. 3-23.

(2) K. H. R. Wright: An Investigation of Fretting Corrosion. Inst. Mech. Eng., proc. B, Vol. 18, No. 11, 1952-53, pp. 556-574.

(3) E. W. Herbek & R. F. Stroheker: Effect of Lubricants in Minimizing Fretting Corrosion. Special Pub. No. 144, ASTM Symposium on Fretting Corrosion, June 1952.

(4) J. M. Bailey & D. Godfrey: Coefficient of Friction and Damage to Contact Area During Early Stages of Fretting; II — Steel, Iron, Iron Oxide & Glass Combinations. NACA TN 3144, 1954.

(5) F. P. Bowden & D. Tabor: The Friction & Lubrication of Solids. Clarendon Press (Oxford), 1950.

(6) H. H. Uhlig, I-Ming Feng, W. D. Tierney & A. McClellan: A Fundamental Investigation of Fretting Corrosion. NACA TN 3029, 1953.

COMMENTARY by B. G. Rightmire (Massachusetts Institute of Technology). The experimental data show that reproducible results are obtained with the present apparatus from use of the wear area of the flat as a criterion of fretting damage. The success of this criterion is ascribable to the geometry of ball on flat, which concentrates the action into a small, well-defined zone.

The use of the area as a damage criterion was not successful in the apparatus of Uhlig and Feng.[1] The criterion there used was weight loss, after removal of oxide by pickling. The geometry of their machine is that of flat on flat.

The present results show that damage under lubricated conditions is markedly less than in the absence of lubricant, and that the debris in the former case contains little, if any, oxide. These observations support the mechanism of fretting previously suggested, in which fretting begins as ordinary wear and then changes to abrasive wear, as the oxide debris accumulates.[2] On the basis of this mechanism it thus appears that the oil acts in three ways: (a) To reduce the initial wear rate; (b) To reduce the formation of oxide debris by restricting the oxygen supply; (c) To aid the movement of wear particles away from the interface.

References

(1) H. H. Uhlig, I-Ming Feng, W. D. Tierney & A. McClellan: A Fundamental Investigation of Fretting Corrosion. NACA TN 3029, Dec. 1953.

(2) I-Ming Feng & B. G. Rightmire: The Mechanism of Fretting. Lubrication Engineering, Vol. 9, June 1953, pp. 134-6, 158-61.

COMMENTARY by J. R. McDowell (Westinghouse Electric Corp.). Fretting corrosion very often takes place in the presence of oil and the author is to be commended for an important contribution to the growing information concerning this troublesome phenomenon.

The author uses a specimen shape which has become common practice both because of its simplicity and practicality, but which unfortunately introduces a variable that effects the application of the results to engineering designs. A one-half inch diameter steel ball on a flat steel surface develops a maximum pressure,

$$q_o = 0.388\ (PE^2/R^2)^{1/3}$$

which is 1½ times the average pressure where P = load on ball, E = Young's modulus of elasticity, and R = radius of ball.

This well-known Hertz equation as presented in Timoshenko[1] also gives the area of contact as 0.00374 sq mm for this case. The author found that with oil present the area increased to something like 0.05 sq mm in 7200 cycles. This would have a pressure of 2840 psi. According to the curves of Fig. 4 the unlubricated specimens reached a wear area in the same running time of the order of 0.20 sq mm, which is 710 psi. H. H. Uhlig, et al[2] showed that fretting increases approximately linearily with pressure. It would follow that the damage to the unlubricated surface would be about 16 times that of the lubricated surface if the pressure were maintained at the higher value. This value is closer to some data the discussor obtained using mineral oil on several combinations of materials as compared to the same materials unlubricated.

There seems to be some discrepancy between the author's data on the effect of humidity and that of both Wright[3] and Uhlig.[2] This might be accounted for in two ways. The number of cycles per test used by the author is much shorter than that used by either Wright or Uhlig. Uhlig used a run in period of 150,000 cycles during which he stated there was a rapid change in the rate of fretting. The discussor also has found this to be true noting particularly that this part of the curve of rate of wear varies more from combination to combination of materials than the rate of wear for a longer time. This rapid initial rate may be affected differently by humidity than the rate for longer periods of time. Uhlig does not show any effect of a maximum wear at 30 per cent humidity, and there are points at both 30 per cent and zero which do not agree with the trend that the author shows. Wright, on the other hand, shows a big drop in rate of wear from zero to 45 per cent humidity and actually a rise in rate of wear at higher than 45 per cent humidity when volume is used as the criterion.

The tests made here are all for a period of testing which is short of the point when "no lubrication" exists according to Fig. 2. This means that the ability of the additive to maintain a film, the main purpose of E.P. additives, has not been checked. It would be very interesting to see if the number of cycles to the point of "no lubrication" for some of the E.P. additives behaves in the same manner as the author has shown very definitely for plain oil in Fig. 3.

It is noteworthy that Almen[4] also found that E.P. additives had no apparent effect on the fretting tendencies of an oil of the same viscosity.

It is unfortunate that tests of a wear of corrosive nature require so much testing time to accumulate much information. The author, in his attempt to avoid the "unlubricated" state, has sacrificed some in obtaining information that could be extrapolated to longer time periods. Without making tests on the additives which have gone into the "unlubricated" state, it seems difficult to agree with the first tabulated conclusion that the effectiveness of a lubricant is fully evaluated, since it is possible that even in the contaminated state an oil with an E.P. additive may form a film more readily and give a longer time to the point where rapid fretting begins.

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AUTHOR'S CLOSURE. Mr. McDowell has applied the Hertz equation to calculate the pressure between the specimens. I do not believe the results are significant after the first half cycle of vibration when we consider that the areas given are of an ellipse, that the geometry of a sphere on a plane is completely destroyed, and that in unlubricated fretting a third phase of debris is found between the specimens. I see that the number of pounds per square inch may decrease with increase in wear spot; but the real area of contact, which is the more important criteria, should remain the same.

The number of cycles to the point of "no lubrication" for mineral oil plus extreme pressure additives does behave similar to that for mineral oil alone. Of course, the slope of that part of the curve before the "no lubrication" point is different for each additive as is the number of cycles to the point. However, the shape of curve after the point is the same in all cases if humidity is held constant.

I see no value in extending tests with extreme pressure additives into the "unlubricated" state because literally there is no lubricant present; it is all soaked up in the debris and far removed from the rubbing zone. It is not a matter of the lubricant being contaminated with wear products as Mr. McDowell suggests. The additive has only one opportunity to reduce wear rate and that is when it can make contact with the rubbing surface.

Personals

R. D. McCormick (above) formerly Publications Engineer for Bell Aircraft Corp., Buffalo, has joined the ASLE National Office Staff as Publications Manager, with offices at 84 E. Randolph St., Chicago 1, Ill. A graduate of Purdue University (B.S., 1953), Mr. McCormick served as Managing Editor of the "Purdue Scientist," and is a member of Sigma Delta Chi and of the Publications Committee for the Society of Technical Writers.

A. F. Brewer, Consultant, former Editor of "Lubrication," and Author of "Basic Lubrication Practice" (Reinhold Publishing Corp., New York, 1955), has been selected to write, compile, and edit the "Lube Lines" column of this and succeeding issues of **Lubrication Engineering.**

Dr. H. E. Mahncke, formerly with Westinghouse Research Labs., has been appointed Assistant Manager of the Research Division of Rayonier, Inc., with offices in Whippany, N. J.

W. M. Murray, Deep Rock Oil Co., has been elected President of the National Lubricating Grease Institute for 1956, with **J. W. Lane,** Socony-Mobil Oil Co., as Vice-President, and **A. J. Daniel,** Battenfeld Grease & Oil Corp., as Treasurer; newly-elected Directors include: **D. P. Clark,** Gulf Oil Corp.; **R. Cubicciotti,** L. Sonneborn Sons, Inc.; **H. P. Ferguson,** Standard Oil Co. of Ohio; **H. L. Hemmingway,** The Pure Oil Co.; **F. R. Hart,** Standard Oil Co. of Calif.; **C. L. Johnson,** Jesco Lubricants Co.; **G. Landis,** The Atlantic Refining Co.; and **H. A. Mayor, Jr.,** Southwest Grease & Oil Co.

I. L. Carmichael has been named President of the Warren Refining & Chemical Co., Cleveland, Ohio, succeeding **B. H. Schulist** who has been appointed Chairman of the Board.

W. Simmons, formerly with Battenfeld Grease & Oil Corp., has been appointed Northeastern Sales Representative for Southwest Grease & Oil Co., Inc., Wichita, Kans.

The Alpha Molykote Corp., Stamford, Conn., announces that **Penn-Petroleum Corp.,** Detroit, Mich., has been appointed exclusive distributor for Molykote lubricants in the Detroit area.

C. W. Mohr, formerly with Quaker Chemical Products Corp., has been appointed Field Project Engineer for Milwaukee Valve Co., subsidiary of A-P Controls Corp., Milwaukee, Wisc.

H. Risko has been named Sales Manager of the Industrial Filtration Division of U. S. Hoffman Machinery Corp., New York City.

Complete lubrication systems will be supplied by Dravo Corp., Pittsburgh, Pa., for a new merchant bar and rod mill (designed by Morgan Construction Co., Worcester, Mass.), for **Atlantic Steel Co.,** Atlanta, Ga.

S. C. M. Ambler has been named Manager of lube oil and grease sales for Canada, for The British American Oil Co., Ltd., Toronto, Canada.

Lubri-Case Inc., has announced the election of **G. B. Troup** as a Vice-President of the company. The New York City concern is the U. S. Patent holding company for the French "Sulfinuz" process.

Acheson Colloids Co. has announced the retirement of **E. A. Lampman** after 25 years of service. **W. H. Leggett** has been appointed Sales Engineer to fill the vacancy in the Rochester, N. Y. sales office.

Dr. J. T. Burwell, Jr. has been named Vice-President of Horizons Inc., an industrial and governmental research organization.

Coming Events

FEBRUARY, 17 & 18, National Society of Professional Engineers (Annual Meeting), Mayflower Hotel, Washington, D. C.

27 thru Mar. 2, American Society for Testing Materials (Committee Week), Hotel Statler, Buffalo, N. Y.

MARCH, 19 & 20, Steel Founders' Society of America (Annual Meeting), Drake Hotel, Chicago, Ill.

APRIL, 4-5-6, American Society of Lubrication Engineers (11th Annual Meeting & Exhibit), Hotel William Penn, Pittsburgh, Pa.

Pumpability Of Aircraft Turbine Lubricants At Low Temperatures

by E. S. Starkman* & J. H. Bridges**

Questions regarding the principles governing flow characteristics of lubricating oils in the vicinity of the pour point have led to an investigation into the behavior of a series of typical aircraft gas turbine lubricants in a simple system at temperatures down to −70 F. The lubricants tested included two oils of military specification MIL-O-6081B (a 1005 grade and a 1010 grade), a sample equivalent in other properties to 1005 grade but having a −15 F. pour point, di(2-ethylhexyl) sebacate, and an SAE 10W motor oil with a −30 F. pour point. Results indicated that laminar flow relationships were satisfactory for predicting flow characteristics down to the pour point so long as proper attention was paid to pump inlet conditions. In addition, there was evidence that fluids with so-called waxy pour points could be satisfactorily pumped to temperatures lower than the pour point, a conclusion which may be of significance.

Nomenclature

d	=	diameter, feet
x	=	distance in direction of flow, feet
L	=	length of conduit, feet
μ	=	coefficient of viscosity, pound seconds per square foot
p	=	pressure, pounds per square foot
Δp	=	pressure drop, pounds per square foot
r	=	radius, feet
R	=	hydraulic radius, feet
τ	=	shear stress, pounds per square foot
v	=	local velocity, feet per second
v_m	=	maximum velocity, feet per second
V	=	average velocity, feet per second
y	=	distance normal to flow, feet

The basic objective of this investigation was to determine, insofar as possible, the flow characteristics for representative gas turbine lubricants in a simple hydrodynamic system in the vicinity of their pour points. The need for such an investigation was made apparent by previous attempts to evaluate[1] and design[2] engine systems, and by an academic interest in obtaining information on the behavior of certain specific hydrocarbon and non-hydrocarbon lubricants at temperatures near their so-called pour point.[3] Some limited information was available in the literature,[4, 5] but this was not felt sufficiently representative of the problem of pumping low viscosity lubricants in aircraft plumbing.

The successful design of any flow system, whether for application to a single turbojet engine, for an integral oil transfer system for multi-engine aircraft,[6] or even for ground handling,[7] depends upon some knowledge of the quantity of energy necessary to move the fluid under the most adverse conditions. This, in turn, depends upon the fundamental laws governing flow, and particularly on the shear stress relationships in the fluid.

The relationship between shear stress and fluid flow for certain systems has been well established for more than 100 years.[8] The specific goal of this investigation was to ascertain experimentally if these relationships were applicable to typical turbine lubricants in a simple but representative flow system under low temperature conditions.

The problem of providing a bearing with satisfactory lubricant at relatively high temperatures and, also, insuring that the lubricant has fluidity at low temperatures, is one which is faced in many engineering applications. In the case of aircraft turbine engines, the problem is specifically that of adequately lubricating the hot turbine bearing with a fluid which must also be pumpable at the low temperatures encountered in Arctic or high altitude operation. The character of the oil for a particular application is fixed by the load requirement at high temperature and, because of the first consideration being that of lubrication, the design of pumps and plumbing becomes completely dependent on the resulting low temperature properties of the lubricant chosen.

In providing an adequate supply system for low temperatures, and particularly for temperatures in the vicinity of the pour point of a lubricant, the question arises as to proper design criteria. A resolution of the problem depends upon whether or not the lubricant acts as a Newtonian fluid, and/or follows laminar flow relationships. Algebraic definition of such conditions is familiar, but will be repeated here to indicate the specific criteria used in making conclusions later.

A Newtonian fluid is one which, by definition,[9] undergoes movement such that the product of its viscosity and velocity gradient is equal to its shear stress anywhere in the flow field. It follows directly, therefore, that Newtonian flow may be expressed as

$$\tau = \mu \frac{dv}{dy} \qquad (1)$$

If the system in which the flow occurs is a horizontal cylindrical flow channel, the pressure gradient may also be simply related to the shear

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stress as follows

$$\frac{dp}{dx} = \frac{\tau}{R} \tag{2}$$

which, when integrated for a flow path of length L, assuming τ and R to be constant, and recognizing the hydraulic radius R to be $r/2$, becomes

$$\tau = \left(\frac{\Delta p}{L}\right)\frac{r}{2} \tag{3}$$

Further, assuming velocity and shear stress distribution to be cosymmetric to the pipe, Eqs. 1 & 3 may be related to give

$$\tau = \mu\frac{dv}{dy} = \left(\frac{\Delta p}{L}\right)\frac{r}{2} \tag{4}$$

and separating variables yields

$$dv = \frac{\Delta p}{\mu L}\,\frac{r}{2}\,dy \tag{5}$$

The velocity distribution for a cylindrical channel can be obtained from Eq. 5 by taking y decreasing and r increasing from the centerline of the cylinder, allowing $v = v_m$ at $r = 0$, and integrating to give

$$v = v_m - \frac{\Delta p}{4\mu L}r^2 \tag{6}$$

Since it can be shown[10] that the velocity distribution in Eq. 6 is parabolic, the average velocity in the pipe is equal to $v_m/2$. With $v = 0$ at $r = d/2$, (the wall), Eq. 6 by direct substitution becomes

$$0 = 2V - \frac{\Delta p}{4\mu L}\,\frac{d^2}{4} \tag{7}$$

and the pressure drop per unit length is

$$\frac{\Delta p}{L} = \frac{32\,\mu V}{d^2} \tag{8}$$

which is the familiar expression for laminar flow pressure drop in a pipe.

The relationship expressed in Eq. 8 was used as the criterion in this investigation in order to establish how closely the flow might correspond to that of a Newtonian fluid, even though, by other definitions, the fluid might be considered non-Newtonian.

APPARATUS. The flow system used in the investigation is shown schematically in Fig. 1. The test section consisted of approximately 40 feet of one inch O.D. by .049 inch wall copper tubing to which pressure taps had been attached 35 feet apart. This test section was of a necessity formed to fit into the limited space of the cold box and, therefore, incorporated 5 return bends of 9½ inch radius. A bend radius to pipe diameter of such large magnitude had little significant influence (less than one per cent) on the resultant pressure drop compared to a straight line of the same tubing.[11] Lubricant was pumped around the closed circuit by a gear pump rated at 300 psi and with a nominal flow rate at 3000 rpm of 3.5 gallons per minute. A variable speed drive allowed operation of the pump over a range of 500 to 3600 rpm, thus providing a 7.2 to 1 ratio of flow rates. The inlet to the pump incorporated a section of two-inch diameter glass. The size served the purpose of reducing inlet pressure drop to a minimum, and the transparency allowed visual observation of conditions at entrance to the pump.

The pump system was enclosed by a cold box designed specifically for the purpose and is shown schematically in Fig. 2. Design of the cold box was based on a need for an inexpensive method for providing temperatures down to less than —80 F. In order to obtain such temperatures economically, it was decided to use dry ice as a cooling medium, placing about 400 pounds on one side of a barrier dividing the box and circulating the resulting low-temperature atmosphere around to the other side of the barrier. A low-speed "transfer" blower was used for this purpose. The cold box thus was in essence a low-velocity "cold tunnel." Temperature control was effected by thermostatically controlled operation of the transfer blower.

The material under test was stored in a supply tank at a slight elevation to the pump. At the end of the system the oil was collected in a measuring tank which was slightly elevated to the supply tank. The system held about fifteen gallons. Flow

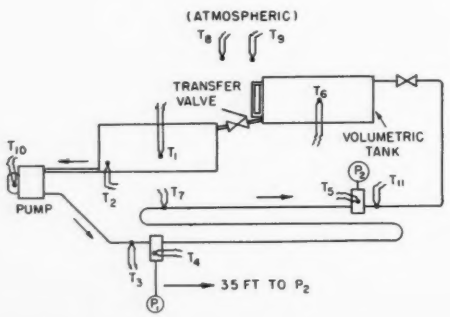

Fig. 1. Schematic layout of system.

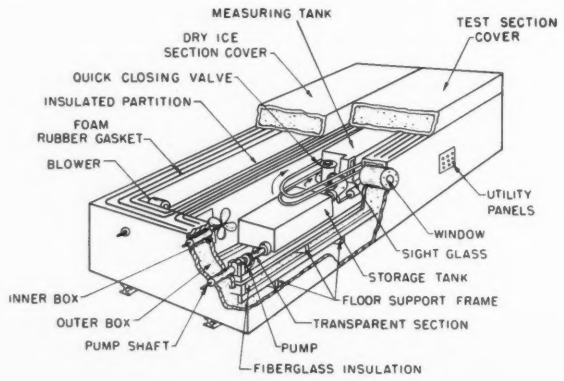

Fig. 2. Cold Box and apparatus.

rate was computed from the measured time for oil to accumulate between calibrated levels in the volumetric tank. During this period the air-operated valve was closed between the two tanks. At the end of a run, the dump valve was opened to allow flow into the supply tank by gravity. During any test the pump was run continuously. The influence of shut-down or soak period was not determined as a part of this study.

The biggest problem to be faced in an investigation of low-temperature pumpability arises from the low mobility of the fluid in the pressure tap lines. Under some conditions these lines may even be at a lower temperature than the portions of the system under test. The speed of response of the pressure-sensing instruments is, therefore, usually unsatisfactory if not altogether impossible.

For this investigation pressure taps and lines were devised as shown in Fig. 3. These pressure taps were made from bakelite and clamped on to the tubing with O-ring seals. Quarter-inch copper lines were inserted into the bakelite to a point just short of the test tubing. These copper pressure lines were then heated by resistance wire to provide fluidity of the lubricant which they contained. However, only enough heat was wanted to allow satisfactory measurements. In order to insure that this heating of the pressure tap lines did not influence the flow characteristics, thermocouples were inserted into the taps at a point near the test section wall, and the current flow to the resistance wire controlled so that the temperature at this point never exceeded the temperature in the flow stream by more than a few degrees. The use of these taps resulted in successful and dependable pressure measurements over the entire range of temperatures encountered.

Thermocouple temperature measurements were taken at the points shown in Fig. 1. The temperature used for relating flow properties of the lubricant is indicated as T_7.

CHOICE OF LUBRICANTS. The selection of test lubricants was based on a compromise in an effort to acquire the maximum amount of information while restricting the number of fluids to a minimum. Five oils were selected which included two representative hydrocarbons and a synthetic, all of military specification, a specially prepared, so-called "hi-pour" 1005 grade (made by a waxy component and a normal component of proper viscosity), and for comparison purposes an SAE 10W motor oil. The first three had pour points of —70 F. or less. The "hi-pour" lube had a pour point of —15 F., and the SAE 10W had a pour point of —30 F.

These five could be listed as: (1) Turbine Lubricant Grade 1010,[12] (2) Turbine Lubricant Grade 1005,[12] (3) Di(2-ethylhexyl) sebacate,[13] (4) SAE 10W motor lubricant, and (5) A specially prepared oil corresponding to 1005 grade in all respects except for a pour point of —15 F.

Pertinent physical properties of the above will be found in Table I, and the viscosities plotted as Fig. 4.

DISCUSSION OF RESULTS. Typical results for one of the lubricants tested, grade 1010, are shown in Fig. 5. Each of the individual flow rates corresponds to a particular pump speed. The gear pump used acted almost as a positive displacement pump for the fluids tested. The top curve, 4.15 gallons per minute, corresponds to 3600 rpm, 2.23 gallons per minute to 1850 rpm, 1.42 gallons per minute to 1200 rpm, and 0.93 gallons per minute to 770 rpm. The solid lines in Fig. 5 were calculated from Eq. 8 and the properties of Table I and Fig. 4. The correspondence of calculated to measured results serves to indicate that at least for this oil the flow was laminar within the limits of measurement.

The region marked as the onset of erratic operation on each of the constant flow rate curves of Fig. 5 corresponds to a combination of difficulties in accurately determining volumetric flow rate, as well as a tendency for the inlet conditions to the pump to become critical. Those points above 5 pounds per square inch pressure drop per foot of line cannot, therefore, be considered dependable. Fortunately, this order of magnitude of pressure

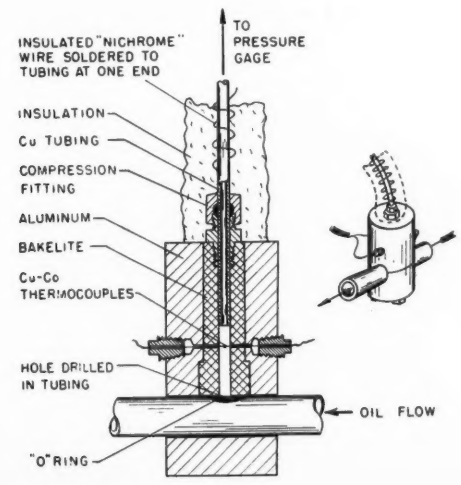

Fig. 3. Heated pressure tap.

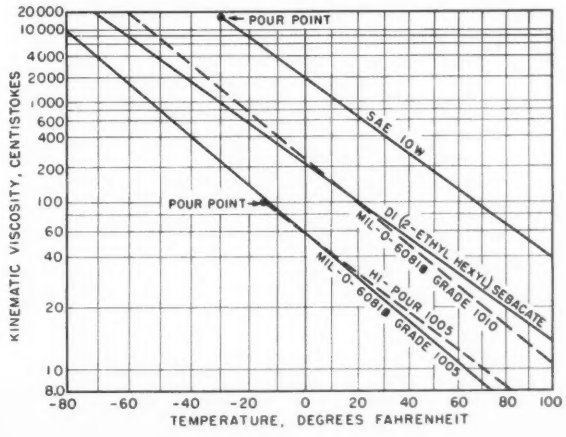

Fig. 4. Lubricant viscosities.

drop is also not highly practical. The data on grade 1010 oil were the first obtained in the investigation. Later tests on the other lubricants incorporated an improved technique, with the result that the divergences in the low-temperature region were not so evident.

Each of the lubricants tested was treated in a manner similar to that described for the grade 1010 oil. For an overall comparison, the results were expressed by a transposition of Eq. 8 to the form

$$\frac{\Delta p}{LV} = \frac{32\,\mu}{d^2} \qquad (9)$$

Since the diameter of the line was a constant, 0.902 inches, the results for each lubricant can thus be expressed as a single curve. These five computed curves are shown in Fig. 6 with the corresponding experimental data. Fig. 6 can then be used for a gross assessment of the phenomenon under study.

Examining the performance of the SAE 10W oil, which had a pour point of —30 F., shows that the flow is, as expected, according to laminar flow relations down to approximately 10 degrees above the pour point. No data could be obtained closer than 10 degrees above the so-called pour point.

The data for the 1010 and 1005 grade oils shown as open points on Fig. 6 gave similar results to the SAE 10W. The 1010 grade was mobile down to about —65 F., and the 1005 grade to —75 F. Even though the cold box was capable of about —90 F., the temperature of —75 F. was the limit for the oil when heated due to pumping. It is evident that both of these hydrocarbon materials acted as expected according to Newtonian flow predictions down to the lowest temperature limitations of the system and apparatus.

Di(2-ethylhexyl) sebacate yielded results as expected over most of the low temperature range. For some reason as yet not clear, the data for the sebacate diverges at —60 degrees. The expected pressure differentials were not obtained, and the magnitude of the discrepancy is believed to be beyond the expected experimental errors. No obvious physical changes were noted through the transparent sections of the flow system. More effort will have to be expended in further investigation of this apparent flow anomaly.

The most interesting behavior was that of the "hi-pour 1005" oil. As is apparent from the closed points on Fig. 6, this fluid was capable of being pumped down to —75 F., well below its —15 F. (nominally —20 F.) pour point. For purposes of comparison, the predicted flow characteristics in the region below the pour points were based on an extrapolation of the higher-temperature (33 F. to 170 F.) viscosity measurements. Obviously, viscosity measurements taken by gravity potentials at the lower temperature would have no meaning, even if obtainable. The data for the "hi-pour 1005" grade oil fall slightly above the predicted curve in the low-temperature range. The significance of this divergence is questionable, since the extrapolated "viscosity" could well have been largely influenced by relatively small errors in the determination of the high temperature viscosity. There is also question that the viscosity-temperature curves in the low-temperature region correspond to straight lines obtained through extrapolation, especially near the pour point.

The behavior of the "hi-pour 1005" oil, while not entirely as expected, is a result of the character of its pour point. Whereas the SAE 10 oil had a

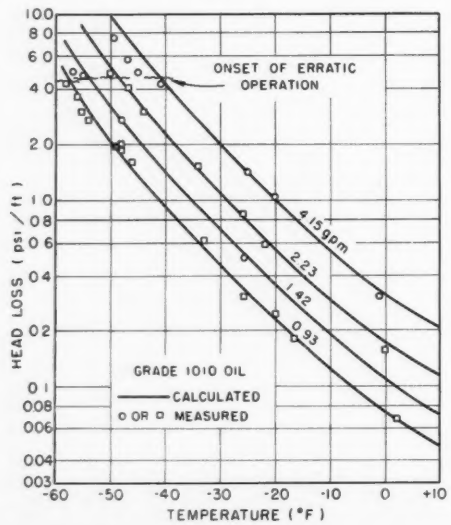

Fig. 5. Flow characteristics for grade 1010 lubricant.

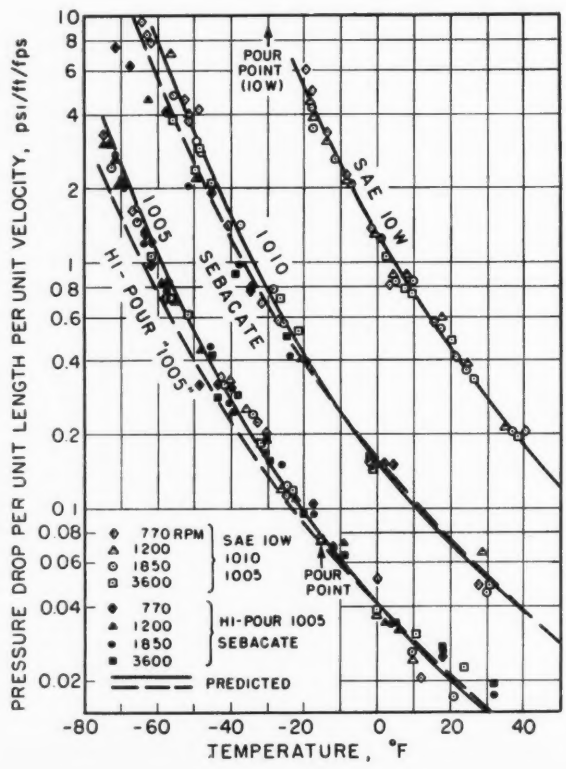

Fig. 6. Combined results.

Table I. Properties data as measured.

Table I Lubricant	Temp. °F.	Viscosity Cs.	Temp. °F.	Specific Gravity	Pour Point °F.
MIL-O-6081B 1005 grade	100	5.23	168	0.822	
	68	8.63	118	0.841	
	-65	2679	80	0.852	
			33	0.871	below -80
			15	0.878	
			-4	0.885	
MIL-0-6081B 1010 grade	100	10.18	60	0.873	
	68	22.4	25	0.886	below -70
	50	36.3	0	0.895	
	-40	2810	-25	0.903	
"Hi-pour" 1005 grade	130	4.41	170	0.822	-15
	100	5.29	90	0.850	
	-10	83.9	61	0.860	
			33	0.871	
SAE 10W	210	6.0	192	0.831	
	130	19.6	137	0.850	
			110	0.858	-30
	100	38.0	80	0.870	
			45	0.882	
Di(2-ethyl-hexyl) sebacate	210	3.3	171	0.873	
	100	13.1	124	0.892	below -70
	-40	1,740	71	0.911	
	-65	10,740	27	0.928	

so-called viscosity pour point, that for the "hi-pour 1005" was a gel and was due to a crystalline structure, composed of a waxy portion of its composition, restraining gravity movement of its remaining components. Once the gel structure was formed and broken, the crystals of which it was composed had little detectable influence on the flow characteristics.

CONCLUSIONS. 1. Pumping of a gas turbine lubricant at temperatures near the pour point presents no difficulty as long as the inlet system is sufficiently generously designed to preclude cavitation at that point.

2. The low-temperature flow characteristics of fluids representative of aircraft turbine applications are as predicted according to laminar flow or Newtonian flow calculations.

3. It is possible under certain conditions to circulate some lubricants at temperatures well below their so-called pour point. These are materials having gel, as contrasted to viscosity, pour points.

4. The flow characteristics below the gel pour point of a representative fluid corresponded to what could be expected from an extrapolation of higher-temperature viscosity.

5. A representative fluid with a viscosity pour point could not be satisfactorily pumped at temperatures within 10 degrees of its pour point. Flow at any condition was according to laminar predictions.

6. Pour point is a poor criterion for predicting the lower limit of pumpability. It has been demonstrated that gel structures which define a pour point have little influence in relation to forced flow. Conversely, viscosity pour points definitely limit the pumpability range.

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ACKNOWLEDGEMENT. Sincerest appreciation is hereby expressed to the Shell Development Co., Shell Oil Co., and the California Research Corp., for their generosity in making available some of the lubricants and properties data which was used in this investigation. Thanks are due to John Randall and Masamichi Watanabe for their aid and perseverance in overcoming operating obstacles and for their help in collecting data.

Lubrication Abstracts

(This is the third in a series of abstracts of translations of Foreign articles on lubrication, by Henry Brutcher. Copies of the full translations are available, at the indicated prices, by writing Mr. Brutcher at P. O. Box 157, Altadena, Calif.)

Boundary-Friction Lubricants (Chiefly for the Cold Working of Metals), by M. Kuhn; "Stahl Und Eisen," Vol. 72, No. 20, 1952, pp. 1212-1216; 2,600-word condensation. Summary of theory of lubrication in hydrodynamic-transitional and boundary lubrication ranges, with special reference to 'surface-bound' lubricants, i.e., lubricants having a capacity for bonding with metal surfaces based on chemical rather than physical forces. Dependence of lubrication on chemical nature of lubricant; mode of action of lubricant carrier. Investigation of exchange action between lubricant and lubricant carrier in wire drawing. Development of theory about the chemical structure of lubricants and the

(Continued on p. 60)

A Progress Report On Silicone Lubricants

by H. F. Lamoreaux*

The first part of this paper is a review of early attempts to improve the lubricating qualities of silicone fluids. Work with additives, and the precoating of bearings, and studies of fluid mixtures are briefly reviewed.

The second part of the paper is a review of recent improvements that have been made in the lubricating qualities of silicone fluids by alterations in their chemical composition. These altered silicones still retain the good qualities of the base fluid in respect to physical properties, hydrolytic stability, and thermostability. An hypothesis is presented to account for the improved lubrication.

Since their commercial introduction some ten years ago, silicone lubricants have presented an enigma to the design engineer. While many times he desired to make use of their almost ideal physical properties, he had to constantly remind himself that these new fluids were poor lubricants for many metal combinations. At times, perhaps, when severe temperature limits were imposed, he could design equipment that operated successfully with silicone fluids. Often this was not practical. The need to develop a silicone fluid having lubricating ability has been a constant challenge.

Early Attempts to Improve Lubricity. Initial attempts to improve the lubrication properties of the silicone fluids were mainly concerned with the use of additives. This was a logical approach. If additives could be found that would give the required lubrication, then the goal could be achieved with essentially no change in the physical properties of the fluid. However, this proved to be a discouraging task. Generally speaking, the silicone fluids responded only mildly when lubricity additives were employed. The problem was further magnified by the fact that the miscibility of most of the additives that were tried was extremely limited in the fluid. The excellent thermal stability of the silicone fluids is well recognized. Few, if any, known additives exhibited the same thermal characteristics. Additives which were soluble at low temperatures boiled out at elevated temperatures. Those which possessed the necessary low vapor pressure to function at the higher temperatures crystallized out as the temperature was lowered. To further complicate the problem, a few additives which answered most of the requirements caused severe metal corrosion at elevated temperatures. It seemed almost impossible to find a single additive which would be compatible over the whole range in which the silicone itself could function.

A report by Brophy, Militz, and Zisman, "Dimethyl-silicone-polymer Fluids & Their Performance Characteristics in Unilaterally Loaded Journal Bearings", had shown that a bearing could be pretreated with a silicone such that it could subsequently operate successfully with a silicone fluid. This treatment involved either heating a bearing in a silicone fluid until a silicone gel was affixed to the bearing surface, or by a long term break-in of the bearing at slowly increased loads while lubricated with the silicone fluid. Bearings treated in this way were able to carry heavy loads at high speeds. This work served to further emphasize the high resistance to shear breakdown of the silicone fluids. The treatment, however, was somewhat involved and of questionable reproducibility.

Another line of attack was to use the silicone to improve the viscosity-temperature characteristics of other fluids having better lubricating ability. Lubrication tests on mixtures of this sort yielded lubricity results which were more nearly those of the silicone, although some improvement could be seen. There was little question, however, that the silicone could improve the thermal properties of those fluids with which it was miscible. Recent work at NACA (Johnson, Bison) has indicated that this approach may have some value. Their data show that mixtures of a silicone and diester fluid have sufficient lubricity while exhibiting a good viscosity-temperature relation.

An extension of this is the use of high viscosity silicone fluids as a polymer thickener or V.I. improver for other fluids or fluid mixtures. Again, its resistance to shear breakdown makes certain that the silicone polymer will continue to function over long periods of time.

Recent Developments. None of these approaches had solved the real problem of creating a silicone fluid which is in itself a good lubricant. It became increasingly evident that the only answer was to create a copolymer in which part of the methyl groups were replaced by a more active substituent. While this substituent must be more surface active, it could not detract from the stability or other desirable physical properties of the silicone fluids. The answer was finally found in the use of chlorophenyl derivatives which appear to fill many of the requirements. This approach has created a new class of silicone fluids which will play an important part as a high temperature lubricant.

In Table I the physical properties of a typical chlorophenyl methyl fluid are compared with its predecessor, a methyl silicone. It should be noted that this fluid is very similar to the methyl silicone in its viscosity-temperature and other physical properties.

The silicone fluids are outstanding for their inertness and resistance to hydrolytic attack. This has been a prime factor in their selection for many applications. These new lubricating fluids have the same order of hydrolytic stability, since no ester or other easily hydrolyzable or acid forming groups are involved. The fluid contains only siloxane linkages throughout the copolymer. Table II gives results from the "stoppered bottle" test.

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Corrosion tests as shown in Table III also attest to the inertness of this fluid.

It is evident that the physical properties of this new fluid have been altered only slightly and compare favorably to the methyl silicones. The similarity ends, however, when the lubrication results are compared on the Shell 4-ball and Falex testers (Table IV). Here the results are equivalent to the more familiar lubricants. A definite increase in the lubricating ability of the silicone fluid has been achieved with no sacrifice of its thermal properties. In fact, its high temperature stability is better than that of a methyl silicone.

While the fluid is not as efficient a lubricant on softer steels, there is sufficient evidence being accumulated to show that this can be improved.

As indicated earlier, the conventional silicone fluids are generally unreceptive to improved lubrication by the use of additives. However, these new fluids respond quite readily to many of the common additives, although some problem of miscibility still exists. While these fluids have increased miscibility with many organic materials, selection of an additive has to be done carefully. Additives can improve the anti-wear properties of these fluids, as indicated in Table V. In each case the decrease in wear is calculated on results of the base fluid with no additive.

These additives would not be suitable for general use in the chlorophenyl methyl silicone fluid. They merely serve to emphasize the point that these new fluids are more responsive to additives.

Why Improved Lubricity? Why do these new fluids lubricate better than previous silicones? There is some evidence which hints that these fluids will react with metals at extremely high temperatures, since a protective film appears to be imparted to metal that has been subjected to high shear loads in presence of the fluid. This is only a supposition, however, and considerable investigation may be required to get a full answer.

One attempt to explore the possibility of a chemical combination was undertaken. Radioactive zinc was heated in the presence of the fluid to a temperature of 350 C. The film of fluid which clung to the metal surface was washed off and checked for radioactivity. None could be found. If there is a chemical activity, then it is above 350 C. Experiments are to be repeated at higher temperatures.

Future Trends. Certainly this new fluid does not represent the ultimate in silicone lubricants. It is, however, a significant step forward. With continued research and development, the lubricity of the silicone fluids will be further improved; especially their ability to lubricate effectively at elevated temperatures. Physical properties will be made even better with added emphasis on high temperature thermal stability.

Along with improvements in the silicone fluids, the introduction of new, inert type grease fillers will make possible the formulation of greases with an exceptionally wide operating range. Certainly one can predict that a range of — 65F. to + 600F. is not an impossibility.

Table I. Physical properties of two silicones. (Key: *Viscosity-temperature coefficient 1-Vis. at 210 F./Vis. at 100 F.; **Average life, measured by testing time to gel in open convection type oven.)

Table I	Methyl Silicone	Chlorophenyl Methyl Silicone
Viscosity -50	500	800
100 F.	40	40
210 F.	17	16
Viscosity-temperature coef.*	.58	.60
Pour Point °F.	-65	-100
Flash Point °F.	599	575
Sp. Gr. 20 C./4	.964	1.03
Thermal stability at 200 C.**	250 hrs	500 hrs.

Table II. Hydrolytic stability, stoppered bottle test + 200 F. for 48 hrs. (Based on requirements of proposed USAF specification for "Fluid Hydraulic, High Temperature Aircraft.")

Table II Property	Requirement	Chlorophenyl Methyl Fluid Results
Copper Wt. Change Mg./sq.cm.	±0.5	0.0
Copper Appearance	No pitting or etching	passes
Acid No. Oil Layer	0.5 (max.)	0.02
Acid No. Water Layer	10 Mg.KOH(Max.)	0.05
% Viscosity Change at 210 F.	-5 to +15%	-0.9
% Insolubles	0.5% (max.)	0.0

Table III. Corrosion test, according to VV-L-791d, Method 530.8.1, 50 hours at 500 F.

Table III Metal	Appearance	Weight loss, gm/cm^2
Copper	Darkened, No etching or pitting	0.0
Silver	Darkened, No etching or pitting	0.0

Table IV. Comparison of lubrication results on the Shell 4-Ball (1 hour at 600 rpm. 52100 steel balls) and Falex Testers.

Table IV	Wear Scar, diameter in MM	
Shell 4-Ball Tester	Steel on Bronze 10 Kg. load	Steel on Steel 50 Kg. load
Methyl Silicone	2.0	1.91
Chlorophenyl Methyl Fluid	0.50	0.55
Petroleum Oil	0.49	0.50
Falex Tester	Ultimate seizure load, No. 8 pins	
Methyl Silicone	300 lbs. gage	
Chlorophenyl Methyl Fluid	1000 lbs. gage	
Petroleum Oil	800 lbs. gage	

Table V. Comparison of wear reduction by additives. (Results listed as % decrease in wear using Falex Tester at 80 pounds gage.)

Table V	Methyl Silicone	Chlorophenyl Methyl Silicone
Additive X	10	100
Additive Y	3 (increase)	24

A Study Of The Sliding Of Metals, With Particular Reference To Atmosphere

by L. F. Coffin, Jr.*

An experimental program is reported on the significance of some of the factors which have been found to affect the sliding characteristics of fairly pure elemental metals. The strength of adhesion of mating asperities, and the resulting frictional force and surface damage, are found to be strongly dependent on the alloying tendency of the metals of the sliding couples. The kind of gaseous atmosphere has a profound effect on local seizure and resultant sliding characteristics of the couples. It has been found that, provided the atmosphere is inert, sliding without local welding occurs for all metallic couples tested which do not alloy.

Nuclear reactors currently being developed for power generation employ fluids, such as water, sodium, or a sodium-potassium eutectic mixture, as principal heat transfer mediums. These fluids must be used at high temperatures and, in the case of water, high pressures in order to achieve reasonable thermal efficiencies, or to prevent formation of a vapor phase within the reactor. Although these fluids are now being used for applications not conceived of a decade ago, one of the problems which arises is age-old but still demands serious attention. The problem, of course, is that of friction.

In nuclear reactors and allied components there are many applications where friction is encountered. Examples include control mechanisms, bearings for mechanical pumps, and bolted connections. For one reason or another it is necessary that surfaces be made to slide when completely immersed in the heat transfer fluid at temperatures up to the maximum encountered in the unit. Unfortunately, both sodium and water are extremely poor lubricants. Sodium, which has been the fluid most widely studied in our Laboratory, is an active liquid metal which effectively cleans metals of their protective films of water vapor and, in some cases, of their oxides. A second difficulty, from a lubrication standpoint, is that temperature and cleanliness considerations prevent the use of lubricants in most applications. A third restriction is that the majority of metals, including practically all conventional bearing metals and alloys, have poor corrosion resistance in sodium.[1] Similar difficulties exist for sliding surfaces in high-temperature, high-pressure water.

Since the boundary conditions imposed on the engineer in bearing design applications for these fluids are highly restrictive, it is immediately apparent that there is little empirical information or scientific fact that can guide him in the proper selection of bearing materials, loads, size, etc., for trouble-free operation. For this reason, the Knolls Atomic Power Laboratory has undertaken a continuing program during the past five years on the basic phenomenon of sliding surfaces in an attempt to learn more about the problem and to aid the designer in his search for satisfactory bearings to apply in the unusual conditions with which he must work. The investigation which is reported here represents a portion of that study.

Materials in Sliding Contact. One would like to follow some reasonable guide in attempting to select proper materials for bearing application where the fluid provides little or no surface protection. The alternative is to undertake an extensive experimental survey where all possible materials are tested two at a time and the results are evaluated according to friction coefficient, wear, and surface damage. This is an exceedingly expensive and time-

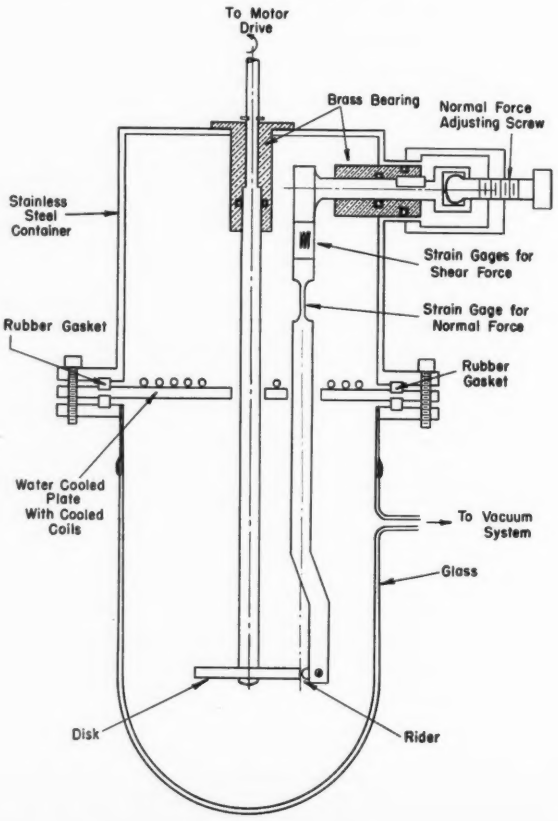

Fig. 1. Boundary friction apparatus.

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(This paper was co-sponsored by the ASLE Technical Committees on Bearings & Bearing Lubrication, and Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

consuming process, although many experiments of this kind have been carried out in various laboratories.

A theory which would seem quite reasonable as a guide in choosing the mating pairs in a sliding metallic couple is to select those pairs which have very low mutual solubilities. Such a point of view is not new. Ernst and Merchant[2] in comparing experimental and calculated friction coefficients were led to the conclusion that metallic couples exhibiting negligible solid solubility at a given temperature generally had anti-scoring properties at that temperature. More recently Roach, Goodzeit and Totta,[3] in studying the performance of bearing metals against steel shafts, found that good bearing metals were those which at their melting points have extremely limited solid solubilities in iron.

This point of view is somewhat at variance with Bowden's concept of friction and surface damage of unlubricated metallic surfaces. From the work of Bowden, Moore and Tabor,[4] and Bowden,[5] local adhesion and cold welding between mating asperities on the sliding surfaces result in friction and metal transfer, regardless of load or speed. Surfaces which have been carefully cleaned and outgassed in high vacuum have been shown to exhibit particularly high friction and surface damage.[6, 7] Frictional resistance is attributed primarily to the shearing of the metallic junctions and to the work of dragging or plowing the surface irregularities of the harder metal through the softer one. In commenting on the relative solubility concept of Ernst and Merchant, Bowden and Tabor pointed out that while mutual solubility might be significant in certain aspects of the sliding process it was not completely satisfactory. This was based on tests carried out on a couple listed by Ernst and Merchant as mutually insoluble at room temperature, namely, copper vs. iron. Here very slow-speed room temperature tests showed marked pickup of the copper on the steel and some pickup of steel on copper.

A recent paper by Machlin and Yankee[9] reports that for sliding of insoluble mating metals on freshly cut surfaces, solid phase welding occurs in some cases but not in others. For example, cadmium was found to weld to iron as well as any other metal tested, while silver did not weld to iron. To interpret these results a criterion for solid phase weldability was proposed based on the comparative values of the work of adhesion to the strength of the weaker component.

The work reported here relates to the behavior of sliding metallic couples which were selected according to their ability to alloy. Although such a selection was not planned at the very beginning of the work, it soon developed from the pattern of the results that a systematic study was warranted.

Disk and Rider Apparatus. For all the tests reported here, a friction measuring device was employed as shown in Fig. 1. A disk 2¼" in diameter, which serves as one of the metals in the couples to be tested, is mounted at the end of a long rotating shaft. The shaft is driven externally by a variable speed motor, and is guided by a long sleeve bearing mounted in an outer stainless steel shell. The other member of the couple is a small hemispheric rider, ¼" in diameter, attached to the end of a long cantilever beam connected at the opposite end to a circular shaft at right angles. This shaft, in turn, passes through a supporting bearing rigidly mounted in the side of the outer shell. The radial position of the

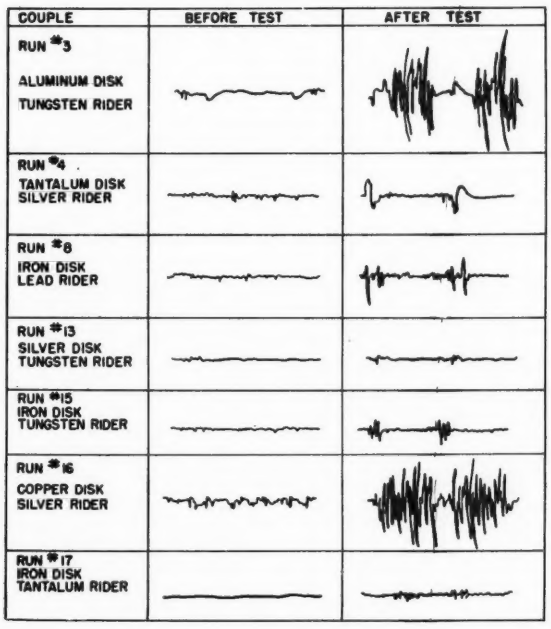

Fig. 2. Brush surface analyzer; traces of wear tracks. Load, 373 gm.; revolutions, 100; vertical magnification, 3740X; horizontal magnification, 31X.

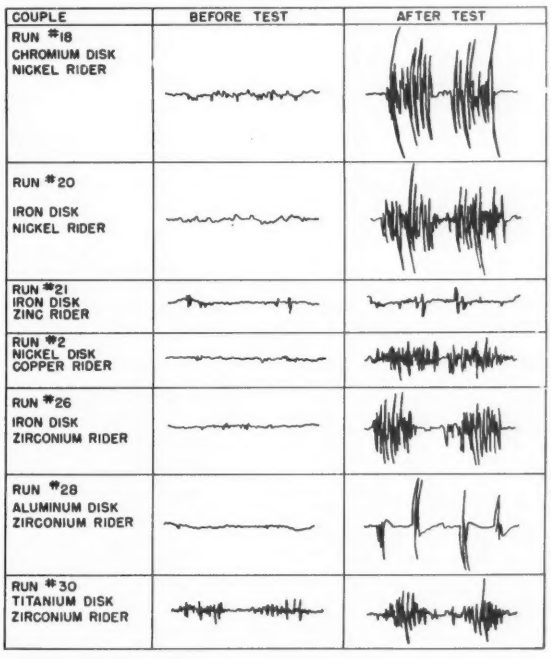

Fig. 3. Brush surface analyzer; traces of wear tracks. Load, 373 gm.; revolutions, 100; vertical magnification, 3740X; horizontal magnification, 31X.

rider can be manipulated externally by a longitudinal motion of the horizontal shaft using the normal force adjusting screw shown. The cantilever beam is supplied with two reduced sections as shown, which serve to amplify bending strains produced either by normal forces or shearing forces on the rider. These bending strains are measured by four SR-4 Bakelite strain gauges in each section to form a complete Wheatstone bridge circuit. Insulation for removal of strain gauge lead wires from the system is provided by Stupakof glass-to-metal seals (not shown).

Physically the apparatus is divided into an upper assembly and a lower assembly. The upper one consists of the stainless steel shell and the disk and rider mechanism. The lower assembly consists of a chamber which is bolted to the upper chamber by means of mating flanges, and is removable to permit access to the disk and rider. This lower chamber can be glass or metal depending on the conditions of the experiment. The complete system is designed to permit a vacuum of 10^{-5} mm. This is accomplished through the use of O-ring shaft seals at the bearings and soft rubber gaskets at the flanged joint.

To prevent the possibility of overheating the strain gauges during outgassing prior to performing vacuum experiments, the two chambers are divided by a water-cooled plate. Two small openings permit the passage of the disk and rider shafts to the lower chamber. When the lower chamber is glass, outgassing can be accomplished by the use of induction heating methods. For prolonged high temperature tests, a thin-walled metallic chamber is used which can be placed in a temperature-controlled heating device. Aside from vacuum, a variety of atmospheres can be employed as warranted by the particular experiment.

The maximum surface velocity of the disk is about 1.8 ft/min, but this can be reduced to as low as desired. The normal load of the rider on the disk generally did not exceed 0.6 lb (273 grams), and in fact was maintained at this value for all tests except as noted. It was impossible to remove completely the eccentricity of the disk so that variation in the magnitude of the normal load occurred with rotation. Since the bulk of the tests were made with many revolutions of the disk, the maximum normal load per revolution was used in the friction calculations. The normal load was monitored by measuring the strain continuously, using an SR-4 portable strain indicator. The shearing force produced by the rider in sliding against the disk is plotted on a Brush recorder, the maximum load per revolution being used in the friction calculations.

Table I. Influence of outgassing on friction coefficient of various metallic couples.

Couple	Degree of Solid Solubility	Air	Coefficient of Friction Vacuum after Outgassing
Cu vs Ni	Complete	0.45	1.50
Ta vs Ni	Complete	.23	.90
W vs Ni	Complete	.21	1.36
Cu vs Fe	Fairly low	.51	.75
Ta vs Cu	Very low	.44	.43
W vs Cu	Very low	.34	.41

Materials and Surface Preparation. A wide variety of materials, including metals, alloys, and ceramics, have been studied. No attempt was made to obtain better than a commercially pure grade for the metals used. In the case of iron, the material used was a 0.10% carbon steel. Disk and rider surfaces were carefully polished to a surface roughness of less than 5 microinches RMS. Degreasing, using both acetone and trichloroethylene, was carried out just prior to performing the experiments.

Early Studies. The first series of tests undertaken was a study of the comparative friction behavior of a variety of metals, alloys, and ceramic materials in air, vacuum, and vacuum following outgassing. Some 75 couples were explored in all. A constant normal load was used and less than one revolution of the disk was permitted. In practically every case there was an increase in the friction coefficient by outgassing in vacua at temperatures up to 800 C. It was further found that the increase in friction varied considerably depending on the couple selected. Although a systematic study in the choice of couples was not performed, there appeared to be a very distinct pattern in the results, partic-

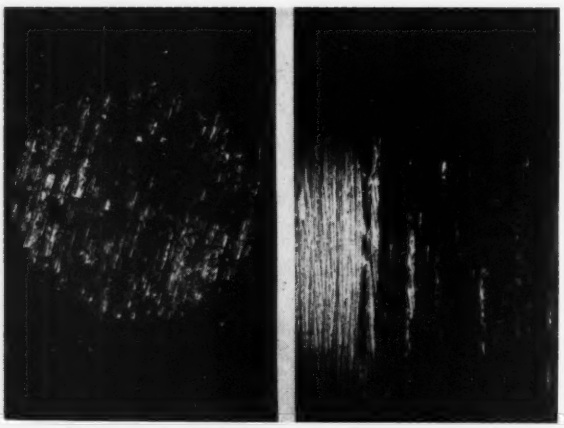

Fig. 4. Run #2; (left) copper rider, (right) nickel disk.

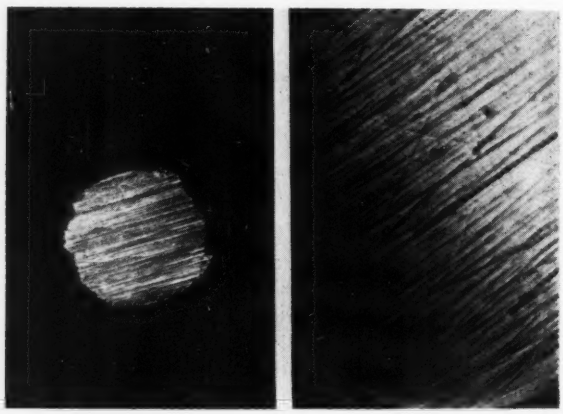

Fig. 5. Run #1; (left) silver rider, (right) iron disk.

Table II. Class A couples with complete solubility; air atmosphere. Key: *Plated; **When rerun for 200 turns friction rose to 1.02.

Table III. Class B couples with solid solubility, with or without intermetallic compound; air atmosphere.

Table IV. Class C couples forming intermetallic compounds, low solubility; air atmosphere. Key: *Increased to 1.02 after 600 turns.

Table V. Class D couples with very low solubility, no intermetallic compounds; air atmosphere. Key: *Plated.

	Run No.	Rider	Disk	Coefficient of Friction After 5 turns	Coefficient of Friction After 100 turns	Appearance of Rider Contact Area	Appearance of Wear Track on Disk
Table II	2	Cu	Ni	0.28	1.00	Abraded, wide	Abraded, wide
	34	Cu	Ni	.29	1.28	Abraded, wide	Abraded, wide
	19	Mo	Cr*	----	----	Abraded, wide	Abraded, wide
	20	Ni	Fe	.82	1.10	Abraded, wide	Abraded, wide
	29	Cr	Fe	.30	.73**	No abrasion, narrow	Some abrasion, narrow
	30	Ti	Zr	.69	.64	Some abrasion, wide	Some abrasion, wide
Table III	16	Ag	Cu	0.45	1.50	Abraded, wide	Abraded, wide
	18	Ni	Cr	.91	1.30	Abraded, wide	Abraded, wide
	38, 15	W	Fe	.19	.54	No abrasion, narrow	Considerable abra., nar.
Table IV	3	W	Al	1.24	1.60	Smeared	Badly abraded, wide
	17	Ta	Fe	.30	.42*	Some abrasion, narrow	Some abrasion, narrow
	21	Zn	Fe	.26	.32	Some abrasion, small	Smeared, narrow
	22	Sn	Cu	1.44	1.63	Some abrasion, large	Smeared, Wide
	23	Sn	Ni	.71	.93	Some abrasion, large	Smeared, wide
	26	Zr	Fe	.39	.72	Some abrasion, large	Pronounced abra., wide
	28	Zr	Al	.45	.72	Smeared	Pronounced abra., wide
Table V	1	Ag	Fe	0.15	0.65	Smeared, medium	None visible
	33	Ag	Fe	.15	.18	Smeared, medium	None Visible
	4	Ag	Ta	.35	.48	Smeared, small	Polished, narrow
	5	Mo	Cu	.29	.60	Polished, small	Polished, narrow
	12	Ag	Cr*	.25	.70	Smeared, medium	None visible
	31	Mg	Fe	.59	1.02	Some abrasion, large	Smeared, wide
	35, 6	Ta	Cu	.15	.25	Some grooves, small	Polished, narrow
	32	Pb	Al	.95	1.30	Some abrasion, large	Smeared, wide
	37	W	Cu	.15	.32	Polished, small	Polished, narrow
	64	Pb	Fe	.38	.93	Some abrasion, large	Smeared, wide

ularly when considering metallic couples. This can be seen in Table I, where the coefficient of friction of various couples is given both in air and vacuum following outgassing. It will be seen that, for the couples listed whose solubility in the solid state is complete, the increase in friction coefficient is at least three-fold, while for those couples with extremely low solubility in the solid state the rise in friction is very slight. When the solubility is fairly low, as in the case of copper vs. iron, the rise in friction is moderate.

The work of Bowden and co-workers[6, 7] indicates that the increase in friction of metallic couples is strongly dependent on surface conditions, particularly when the adsorbed films are removed by outgassing under high vacuum. However, their work gave no indication that the metallurgical relationship of the couples employed was important. It seemed plausible, on the basis of our preliminary results, that the sliding behavior could be different when the degree of solubility was low, and the matter should therefore be investigated. Since there was some question as to how to relate the increase in coefficient of friction from outgassing in studying the comparative surface damage of sliding metallic couples, it was decided to change the test procedure to one in which the friction was measured under conditions of repetitive sliding in an air atmosphere without prior outgassing.

Repetitive Sliding Tests. In planning the repetitive sliding tests it was felt that if the degree of surface damage depended upon the alloying tendencies of the mating couples, then repeated sliding on the same track would lead to damage to the wear track that would be cumulative, and relative differences in damage would be enhanced. These series of tests were all run in air at room temperature.

Couples to be investigated were selected according to their ability to alloy. Four broad classifications were established to identify the alloying tendencies of the couples as follows:

Class A, those couples having complete solid solubility.

Class B, those couples having some solid solubility, and which might or might not also form intermetallic compounds.

Class C, those couples which exhibit a low solid solubility (of the higher melting point metal in the lower melting point metal), but which form intermetallic compounds.

Class D, those couples which do not alloy. The criterion for this class is very low solubility when the lower melting point component is in the liquid state.

Tables II to V list the various couples tested, classified by the above identification. In classifying a particular couple, the equilibrium phase diagram was used where the source for this information was Hansen[10] and the Metals Handbook.[11] It might be argued that the alloy compositions formed at the juncture of mating asperities are not in metallurgical equilibrium. However, the equilibrium diagram certainly serves as a qualitative guide for the alloying behavior of the couple. In cases where solid solubility is indicated in the phase diagram, the solubility of the higher melting-point metal in the low-

er melting-point metal was used, as found from the phase diagram at the eutectic or eutectoid temperature. This again is arbitrary, since the degree of solubility indicated at the higher melting point end of the phase diagram may be the important factor. However, until greater refinements in the understanding of the adhesion mechanism are developed, the above definition will be used.

The number of couples available for this kind of study is extremely large. It is possible to write down a list of over 500 possible test couples using only the fairly common metallic elements. Thus the 24 couples listed in Tables II to V are far from exhaustive. It is felt, however, that a sufficient number of couples have been explored to indicate certain very definite effects.

The test procedure for these tests was to apply a normal load of 373 grams (because of eccentric rotation of the disk the normal load would vary with rotation; as a result, the maximum load was set at 373 grams) and to record the shearing force. The test was continued for 100 revolutions of the disk. The coefficient of friction could thus be determined as a function of the number of turns. Hence, the shape and magnitude of this data, when plotted, serve as one measure of the damage of a particular couple. In addition, the surface roughness of the disk normal to the wear track both before and after the tests was used as a basis for damage. Typical values are seen in Figs. 2 and 3. Finally, the contact area of the rider and the wear track of the disk were photographed for visual observation of the effect of sliding. Figs. 4 and 5 show the results of the sliding of typical couples.

A careful study of the tables and figures reveals that the alloying ability of the couples plays a significant role in the friction and surface damage. For the group designated as Class A considerable seizure generally occurs, resulting in a rapid increase in the friction coefficient and a wide, abraded contact area and wear track, and evidence of considerable metal transfer. The copper-nickel couple (Fig. 4) and the nickel-iron couple are good examples of this kind of behavior. Quantitative results are given in Table II.

The width of the wear track varies from couple to couple, and seems to be a function of the hardness of the rider. Chromium was by far the hardest material of the riders tested, being file hard, and this property is reflected in the resultant contact area of the chromium rider and wear track of the iron. For this couple, abrasion is found only in the wear track of the disk.

The coefficients of friction of the Class A couples in general rise to quite high values after a few revolutions. Coefficients of friction recorded are average values and do not reflect the momentary high shearing forces developed for each seizure. In the case of the chromium-iron couple the increase in friction was much less rapid and exceeded 1.0 only after 200 turns.

Those couples designated as Class B and having somewhat limited solubility are listed together with their sliding characteristics in Table III. Comparative surface roughnesses are shown in Figs. 2 and 3. Behavior of these couples in general is similar to the completely soluble couples of Table II. The contact areas and wear tracks are large and abraded; the coefficients of friction rise sharply with the number of turns reflecting the degree of seizure and surface damage which occurs. Hardness appears to play a role in the degree of damage, as seen in the case of the tungsten-iron couple (Table III). Here the hard tungsten rider is effective in reducing the area of damage in both rider and disk, and the rate of rise in friction.

The Class C alloys, i.e., those which have low solubility but which form intermetallic compounds according to the equilibrium phase diagram, are listed in Table IV. Comparative surface rough-

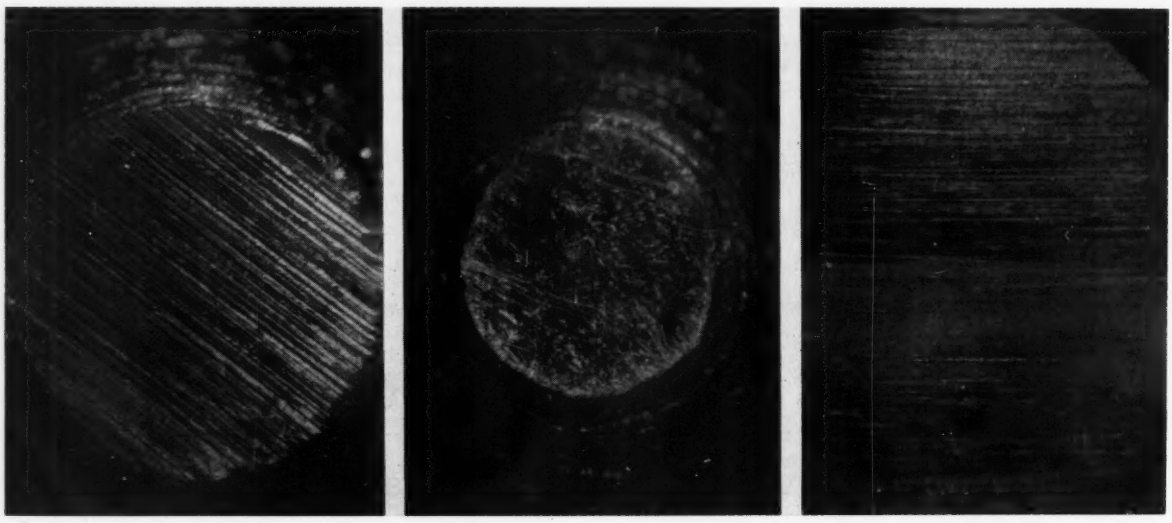

Fig. 6. (left) Lead rider run in dry air; (center) Lead rider run in dry helium; (right) Iron disk, (top portion) wear track after 300 revolutions in dry helium atmosphere, (center portion) wear track after 300 revolutions in dry air atmosphere. 20X.

Table VI. Influence of atmosphere on sliding characteristics.

Rider	Disk	Alloying Classification	Atmosphere	Coefficient of Friction After 5 turns	Coefficient of Friction After 100 turns	Appearance of Rider Contact Area	Appearance of Wear Track on Disk
Cu	Ni	A	Air	0.28	1.00	Abraded, wide	Abraded, wide
Cu	Ni	A	Helium	.58	5.25	Badly abraded, wide	Abraded, wide
Ti	Zr	A	Air	.69	.64	Some abrasion, wide	Some abrasion, wide
Ti	Zr	A	Helium	.92	1.23	Some abrasion, wide	Some abrasion, wide
W	Al	C	Air	1.24	1.60	Smeared	Badly abraded, wide
W	Al	C	Helium	1.05	1.31	Smeared	Badly abraded, wide
Sn	Cu	C	Air	1.44	1.63	Some abrasion, large	Smeared, wide
Sn	Cu	C	Helium	.48	1.00	Slight abra., large	Some smearing, med.
Pl	Al	D	Air	.95	1.30	Some abrasion, large	Smeared, wide
Pl	Al	D	Helium	.50	.49	Medium, polished	None visible
Mg	Fe	D	Air	.59	1.02	Some abrasion, large	Smeared, wide
Mg	Fe	D	Helium	.45	.62	Medium, polished	Narrow, polished
W	Cu	D	Air	0.15	0.32	Polished, small	Polished, narrow
W	Cu	D	Helium	.42	.60	Polished, small	Polished, narrow
Pl	Fe	D	Air	.38	.93	Some abrasion, large	Smeared, wide
Pl	Fe	D	Helium	.38	.42	Polished, medium	None visible
Cd	Fe	D	Air	.39	1.45	Abraded, large	Badly smeared, wide
Cd	Fe	D	Helium	.32	.36	Polished, small	None visible

nesses are given in Figs. 2 and 3. One would expect quite variable results from this group since, if intermetallic compounds are formed, the comparative strength of these compounds and the base metals would determine where the fracture of the adherent contact points occurs. This indeed appears to be the case. For those pairs where one of the metals was quite soft, such as aluminum in the aluminum-tungsten couple or tin in the tin-copper couple, the softer metal was smeared onto the harder. Damage in these cases is severe and the friction coefficient quite high.

There are two cases, that of zinc vs. iron and tantalum vs. iron, where the interface strength appears to be weaker than the base metal. Here the damage on the wear track is slight, the contact area on the rider small, and the friction coefficient low. For the tantalum-iron couple, the test was extended to 600 turns, but the resulting damage did not appear to be any more severe. The friction coefficient, as noted, rose to 1.02 after 600 turns.

Finally, the Class D couples, those which do not alloy, are listed in Table V together with their sliding characteristics. Here a pronounced difference in the sliding behavior is found in most cases from those of the other groups. In the case of silver vs. iron (Fig. 5) and silver vs. chromium, where the rider is much softer than the disk, the silver rider is smeared and polished and no wear track is observed on the disk. Where the relative hardnesses of the disk and rider are the same or where the rider is harder, such as molybdenum vs. copper, a narrow wear track with highly polished grooves is found together with a small, highly polished contact area on the rider. In these cases the friction rise is small and the friction coefficients remain at a comparatively low level after 100 revolutions. For these couples there is no evidence of abrasion, and the friction records give no evidence of local seizure or welding.

It will be noted in studying Table V that not all the non-alloying couples tested behave in the manner described above. Couples whose riders were very soft, such as lead vs. iron, magnesium vs. iron, and lead vs. aluminum, gave a pronounced smearing onto the disk and a large abraded contact area on the rider. Further, the coefficients of friction tended to be high.

Except for the results of the above three tests, the alloying hypothesis for adhesion and surface damage in sliding contact appeared to have been substantiated. Where alloying was indicated from the equilibrium phase diagram, damage was pronounced, welding of mating asperities occurred, and friction was high. Where no alloying was indicated by the phase diagram, seizure did not occur, and the riders and disks, although flattened or grooved by the differential hardness of the couple, had a polished appearance on the contact area or wear track. It appeared that the alloying concept was important in metallic sliding, but that some other factor or factors were also important since three of the non-alloying couples did not behave properly. As a result, a further series of tests was undertaken on the lead-iron couple to attempt to understand this anomalous behavior.

Tests on the Lead vs. Iron Couple. It was first thought that surface roughness of the iron disk was important in the smearing of the lead onto the iron, since the filing action of a rough surface against a soft rider might produce a base for the lead to build upon with repeated sliding. However, a careful polishing of the disk to produce a surface finish of less than one microinch RMS roughness made little difference upon the smearing and friction behavior. It was then reasoned that a low melting point metal such as lead might possibly melt during sliding and adhere to the iron surface by a wetting action. As a result, low temperature tests were carried out down to liquid nitrogen temperatures in air to reduce this possible effect. Even at a temperature of —196 C. on highly polished disks, lead was observed to smear onto the disk, although the degree of smearing and rise in friction were considerably reduced. It was observed, however, that when the couple was completely immersed in liquid nitrogen, there was no smearing and friction remained at a low level during the test. This would seem to indi-

cate either that the kind of atmosphere was important or that a substantially lower contact temperature was maintained through the use of a liquid medium.

Tests were therefore undertaken to study the effect of atmosphere. Dry helium was introduced to the apparatus, and after sufficient purging the couple was run for 300 revolutions at room temperature. The same experiment was repeated in dry air using a new lead rider rubbing against another portion of the disk. The differences are most striking, as seen in Figure 6. It is immediately apparent that, whereas smearing on the disk and wearing away of the rider takes place in air, in helium there is no smearing. The contact area of the rider is polished and is considerably smaller. The difference in behavior is reflected in the coefficient of friction as seen in Fig. 7. In helium the rise in friction is slight, while in air the friction coefficient increases rapidly and levels off at a very high value.

Because the differences in sliding characteristics were so striking, other atmospheres were studied using a new rider and disk location. Results are shown in Fig. 8. Following 60 revolutions in helium, where the friction coefficient reached a constant value of 0.39, dry nitrogen was introduced causing a friction rise to 0.48 after 120 total turns. Dry helium was reintroduced, dropping the coefficient of friction to 0.42 after 85 additional turns. Dry oxygen then caused a drastic rise to 0.76 after 25 turns. The system was then evacuated and hydrogen was admitted. The friction coefficient then levelled off at 0.60 following 280 total turns. Reintroduction of helium only slightly reduced the friction coefficient to 0.54.

From these tests it can be concluded that the smearing of lead on iron is strongly dependent on the atmosphere. The alloying hypothesis is confirmed when the atmosphere is inert helium. It would appear that the smearing found when oxygen is present is due to some interaction between lead, iron, and their oxides formed as a result of a temperature elevation of the metals at the various points of contact. The detailed mechanism has not been explored, however. An explanation for the slight increase in friction of the couple in a nitrogen atmosphere is more obscure, but possibly can be related to the purity of the nitrogen.

Sliding Characteristics of Various Couples in An Inert Atmosphere. Because of the pronounced sensitivity of the lead-iron couple to atmosphere and because all the other couples were tested in air, it was necessary to reexamine the sliding characteristics of the various couples in an inert atmosphere before any definitive conclusions regarding the alloying hypothesis could be drawn. Therefore, couples from each alloying classification were selected and the tests repeated in an inert helium atmosphere. Table VI shows the results of these experiments. It will be seen that in most cases the inert atmosphere acts to accentuate the sliding characteristics expected for that couple. For those couples bearing the "A" alloying classification, particularly in the case of the titanium-zirconium couple, the friction coefficient increases substantially in helium but the visible surface damage is about the same as in air.

The effect of helium on the couples which form intermetallic compounds, Class C, appears to depend on the couple. For example, the tungsten-aluminum couple is changed very little from the severe damage suffered in air, while the tin-copper couple is considerably improved. Smearing of the tin onto the copper does still occur but is considerably reduced, suggesting that oxide formation and interaction affects the adhesion of mating asperities for this couple.

When the non-alloying couples, Class D, are tested in a helium atmosphere, the results are most striking. For those couples which smeared badly

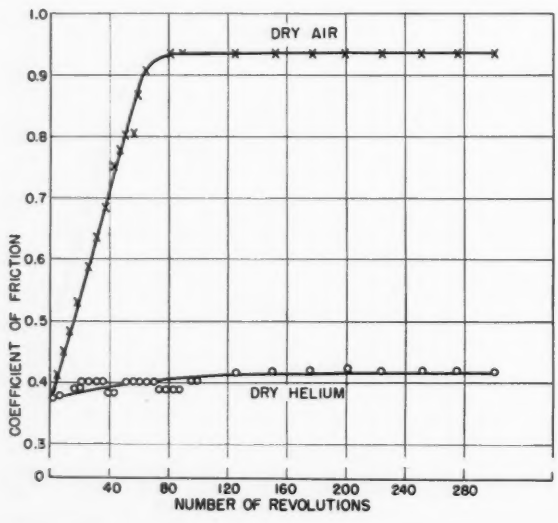

Fig. 7. Effect of atmosphere on friction coefficient of lead-iron couple.

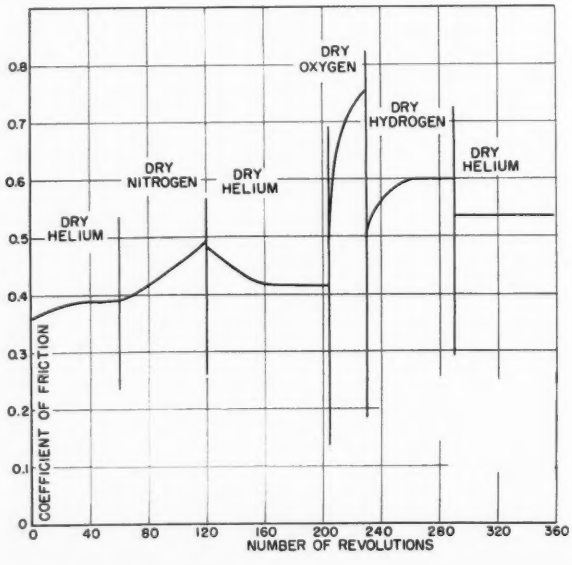

Fig. 8. Effect of change in atmosphere on friction coefficient of lead-iron couple.

and developed a high coefficient of friction in air, now there is no smearing and the friction coefficients remain at low levels. This is seen in Table VI for the lead-aluminum and magnesium-iron couples. The behavior of these couples appears to be identical with that of the lead-iron couple as shown in Fig. 7. For the non-alloying couples which slide without seizure in air, no change is observed in the wear track or contact area when a helium atmosphere is used. However, an increase in friction is found. The tungsten-copper couple is an example of this.

It is thus seen that experimental evidence in support of the alloying hypothesis for seizure and surface damage is strengthened when care is taken to eliminate the important effects of atmosphere. In the presence of an inert atmosphere the interaction between mating asperities is related to the metals themselves, since the initial surface films are removed or broken up by repeated sliding and new films are prevented from forming.

Discussion of Results. From the results obtained on the sliding characteristics of metallic couples when classified according to their alloying ability, it appears that two distinct modes of sliding exist. The first mode is that which takes place with non-alloying couples such as those listed in Table V. This may be described as sliding by shear, where the adhesion between mating surfaces is weak and where there is no other visible disturbance except for conforming of the mating metals by plastic deformation as governed by the gross mechanical properties of the rider and disk. It is, of course, possible to cause metal transfer in non-alloying couples by a filing action, where the surface of the harder metal is rough and where the softer metal accumulates in the recesses of the harder metal. This behavior is also mechanical in nature and can be reduced or eliminated by care in using smooth surfaces.

The second mode of sliding is that which occurs for couples which alloy. Here sliding by shear occurs accompanied by local welding. This welding causes transfer of metal, the abrasion of the sliding surfaces, and high friction. The amount of welding which occurs depends on the couples involved, and factors such as solid solubility, intermetallic compound formation, relative hardnesses, and strengths enter into the problem.

Atmosphere plays an important role in the sliding process. The behavior of oxygen can be explained readily, since the shearing action which occurs in sliding causes the exposure of fresh metallic surfaces upon which new oxide films are formed. Most oxides will interact,[12] and it is presumably this interaction which causes smearing of the soft metal riders when rubbed against non-alloying metallic disks. Smearing should be most pronounced in the softer metals where the strength of the adhesion of the oxides can exceed the low shear strength of the soft base metal. The role of nitrogen in increasing the friction of the lead-iron couple is obscure.

One might speculate as to the role which the alloying ability actually plays in causing local welding. The temperature rise and extreme pressures existing at contact points have been thoroughly investigated by Bowden and his co-workers.[13] It might be expected that surface atoms will be extremely mobile, so that under conditions of intimate contact mutual accommodation of atoms from the two surfaces can occur under proper conditions. This would be apt to occur for atoms which are of similar size or which form chemical bonds, while for atoms which are different in size and do not bond to form an intermetallic compound, accommodation would be more difficult. This accommodation tends to reduce the discontinuity in structure which would otherwise exist, and it would be expected under such conditions that the interface would develop considerable mechanical strength. The accommodation of the mating surfaces would undoubtedly bear some relation to the solid work of adhesion for the two materials, if such could be measured Since the properties which contribute to the interfacial accommodation are similar to those which lead to alloying, it is not surprising that the alloying ability is the important criterion in determining which of the two modes of sliding occurs.

In comparing the present results with the work of Bowden,[4, 5] it has been found here that sliding can occur without welding provided the metals do not alloy (the Cu-Fe couple discussed earlier would not be considered a non-alloying couple by the definitions of the present paper) and an inert atmosphere is used. It should be pointed out, however, that even under these conditions metal transfer still can occur either by mechanical action (filing) or by adhesion. In the latter case, the cohesive strength of surface asperities may be weakened by severe plastic flow or fatigue, so that the adhesive forces are sufficient to overcome the cohesive effects of the softer metal, and material transfer takes place. However, subsequent sliding will cause these particles to be removed or re-welded to the original surface, so that there is no cumulative damage to the wear track. This is actually observed for the softer riders of non-alloying couples, where it is found that material accumulates at the leading edge of the contact area of the rider.

The results reported here are in contradiction to the theory developed by Feng,[14] who proposed that metal transfer and wear are due principally to the mechanical interlocking of roughened interfaces. The pronounced difference in sliding characteristics of the various couples, such as lead-iron, magnesium-iron, and lead-aluminum in air and in helium, with no change other than atmosphere, is impossible to account for from a mechanical point of view.

The relationship between the results of the present investigation and those of Machlin and Yankee[9] should also be considered. They concluded that a ratio of work of adhesion to the strength of the weaker component was the proper measure of solid phase weldability in sliding metal contact. Evidence for this concept stemmed from tests on certain mutually insoluble couples, principally cadmium vs. iron, in which it was found that welding of the cadmium to the iron occurred as readily as

metal riders which were soluble in iron.

The cadmium-iron couple was not on the list of non-alloying couples investigated in the original program at KAPL. However, to verify the behavior of the couple particularly as to atmospheric effects, a comparative run was performed recently of a cadmium rider vs. an iron disk both in air and helium. The results are shown in Table VI. The behavior of this couple was quite similar to the other non-alloying couples with soft riders in that smearing on the disk, abrasion and a wide contact area on the rider and high friction were produced by sliding in air. In helium the rider exhibited a small polished contact area, the disk showed no visible wear track, and the coefficient of friction remained low. The helium test was continued until a total of 300 revolutions was reached with no visible change other than a slight increase in friction to 0.38.

One point should be brought out in connection with the tests of soft riders and non-alloying mating disks in inert atmospheres. Since no attempt has been made in the experiments to remove adherent oxide films, it may be that the soft metals such as lead, cadmium, and magnesium do not penetrate the surface films of the disk. Hence, sliding may be predominantly between the soft metal and the oxide of the harder disk material. This is not the case when the two metals are of similar hardness or when the rider is harder than the disk, since here the oxide film is rapidly broken up and penetrated, and direct metal-to-metal contact occurs.

Despite the possible contradiction in the sliding characteristics of the cadmium-iron couple existing in Machlin and Yankee's tests and those reported here, there appears to be no basic disagreement in interpretation of the two investigations. Actually a criterion based on the ratio of work of adhesion to strength of the weaker component is perhaps more general, and certainly can be better adopted to quantitative expression than a criterion based on alloying ability. It will, however, be difficult to apply at present because of the lack of information on surface-free energies, and interfacial free energies in the solid state, and because of lack of an exact definition of the strength of the weaker component. The non-alloying criterion, on the other hand, is qualitative and probably less general in nature, but experimental evidence has, to date, lent support to its use.

Conclusions. The following conclusions may be drawn, based on disk and rider tests on metallic couples in various atmospheres under conditions of repetitive sliding:

1. The alloying ability of metallic couples serves as a qualitative criterion for local seizure and surface damage.

2. Couples which do not alloy slide without local seizure, as evidenced by the lack of abrasion and by low friction.

3. The sliding characteristics of metallic couples are strongly influenced by the surrounding gaseous atmosphere.

4. An inert gas atmosphere such as helium is required to produce sliding without local seizure for non-alloying couples, where one component is a soft metal.

5. Complete removal of oxide films from the harder metal of non-alloying couples in an inert gas atmosphere may affect the sliding characteristics of metallic couples in an inert gas, if a substantial difference in hardness exists between the components of the couple.

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COMMENTARY by E. S. Machlin (Columbia University). The criteria for local welding during sliding used by Dr. Coffin are the effect of an inert atmosphere on the coefficient of friction, the surface roughness normal to the wear track, the width of the wear track, observation of metal transfer, and the rapidity in rise of coefficient of friction with successive sliding over the same track. It is interesting to note that in our experiments, in which an entirely different set of criteria for welding was used, somewhat similar conclusions have been reached. Our set of criteria are based on comparing the friction coefficients of as-freshly cut surfaces in an inert atmosphere, for the same rider against different slider metals and vice-versa, and also on noting the effect of introducing air, of varying the hardness, and of varying the temperature on the friction coefficients. As a consequence of this series of measurements we have cata-

logued the couples into three groups: completely weldable, partially weldable, and of limited weldability. This list is attached. It is to be noted that, insofar as those couples which have been classified completely weldable, there is no disagreement between the conclusions of Dr. Coffin and myself. In the group which I have classified "weldability limited" there is again no disagreement between the two investigations. However, some of the couples which I consider partially weldable, namely Cu-Ag, Zn-Fe, Fe-Zr, Al-Zr, Dr. Coffin classifies in his classes B and C as weldable, and two others (Al-Pb and Cd-Fe) he classifies in his group D as non-weldable. I am, therefore, mainly in agreement with his conclusions, and to a minor extent in disagreement with respect to the classification of the Pb-Al and Cd-Fe couples. I want to emphasize the agreement rather than the disagreement because it seems to me to justify the basic conclusion we have made together, namely, that all metals do not necessarily solid weld to each other.

Room Temperature Weldability Of Dissimilar Metals

Completely Weldable	Partially Weldable ($\alpha \simeq 1/2$)	Weldability Limited
Pb-Cu	Cu-(com.)Ti	Ag-Zr
Cu-(Iodide Ti)	Cu-(com.)Zn	Pb-Fe**
Zn-Cu	Cu-Ag	Ag-Fe
Zn-Zr	Zn-Fe	Mg-Fe**
Cd-Cu	Zn-Ti	Cd-Zr**
Al-Ag	Zn-Ag	
Al-Cu	Cd-Fe	
Cu-Fe	Al-Fe	
Mg-Cd	Al-Ti*	
Mg-Al	Ag-Cu	
Mg-Cu	Fe-Ti	
Mg-Ti	Fe-Zr	
Cd-Ag	Bi-Fe	
Cd-Ti	Ag-Ti	
	Pb-Al	
	Zn-Al	
	Mg-Ag	
	Al-Zr	
	Al-Ni	
	Bi-Ti	
	Bi-Cu	
	Pb-Ti	

*may belong in 1st group; **may belong in 2nd group

I can offer a number of suggestions for our different classifications of the Pb-Al and Cd-Fe couples. One has already been recognized by Dr. Coffin, namely, that even in the helium atmosphere the contact may be between clean rider and contaminated slider. The drop-in friction coefficient due to cleaning the atmosphere from air to helium may arise as follows: If the oxide developed on one of the components is harder than the underlying substrate, then the mating couple may penetrate the oxide film. This film then breaks up, embeds in the soft matrix, leaving hard cutting edges extending outwards from the matrix. These oxide cutting tools then act to provide a high coefficient of friction. If the oxides are eliminated by running in helium, then their contribution to the friction coefficient will likewise be eliminated. I prefer my own classification of these couples.

With respect to Dr. Coffin's criterion for alloying tendency, I should like to point out that it is not unambiguous. Atoms which differ widely in size do not like to dissolve in a solvent of the other, not only in the solid state but in the liquid state at not too high temperatures as well. Yet the attraction between the atoms may be quite high. It is my belief that this attractive force is basically the cause of contact welding. I am hesitant to use Dr. Coffin's criterion because we have found that not only does the Ag-Fe friction coefficient increase markedly with temperature, but also liquid silver wets solid iron under hydrogen (contact angle about 50°). This, despite the fact that it does not alloy in Dr. Coffin's sense of the term. I therefore suggest that it is simpler to run friction tests to determine weldability than to predict it from first principles.

AUTHOR'S CLOSURE. The author appreciates Professor Machlin's discussion of the paper. It is gratifying to note that there is general agreement in the interpretation of the basic observations, namely, that welding is not a universal property of all metals in sliding contact. This important fact must be recognized in any theory of mechanism for adhesion of sliding surfaces.

Professor Machlin raises a very interesting point in referring to the temperature dependence of the friction coefficient. Unquestionably, one who is interested in studying the friction of metals as a problem in surface phenomenon would attempt to define the behavior in terms of the pertinent physical quantities, namely, work of adhesion, contact angle, interfacial free energy, etc. From this point of view a description of the degree of surface damage in terms of the indicated alloying from the phase diagram is crude and qualitative. The definition of the present paper is not intended necessarily to satisfy the scientist, but rather the more applied man whose knowledge of the subject of surface behavior is limited and who needs some working rule to go by in choosing metallic couples. When one is concerned with such factors as sliding velocity and temperature, unquestionably a mechanism is needed which can be expressed in a more quantitative fashion. From this point of view a criterion of damage such as given by Professor Machlin is more acceptable.

The disagreement in some of the results of the present paper and those in Professor Machlin's studies, particularly for the couples Pb-Al and Cd-Fe, possibly can be explained from geometrical considerations. In our tests these couples were classified as non-alloying and did not undergo local seizure and welding under repeated sliding in an inert atmosphere, while Professor Machlin has found these couples to be partially weldable (see his table). Our results were obtained for the case in which the softer metal was the rider. One of these tests has recently been reversed, i.e., the harder metal was the rider. In contrast to the results previously obtained for the Pb-Fe couple, the friction gradually increased to very high values upon repeated rotation, and lead was transferred to the iron rider. The rate of damage was sensitive to atmosphere; that is, damage occurred much more rapidly in air than in helium. Nevertheless, in helium the sliding characteristics were quite unlike those of the original couple, where the friction coefficient remained low and no adhesion or surface damage was observed.

It would appear, then, that the geometry of the system affects the welding picture. Where a comparatively weak metal is one of a mating and non-alloying pair of metals, the strength of adhesion may be sufficient to remove particles of the softer metal to the harder one. When the softer metal is the rider, this particle can in turn be retransferred because of the plowing force of the rider against the particle. On the other hand when the harder metal is the rider, particles of the softer metal are trapped and accumulate on the leading edge of the rider and can act to interfere with normal sliding behavior. The result then is an increased friction coefficient. Hence, with those cases where one of the metals is very weak, the configuration of the experiment may determine whether high friction, welding, and metal transfer can take place.

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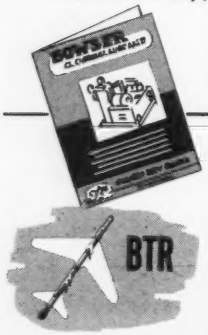

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(Section News, from p. 23)

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November-December. "Additives," by Dr. H. H. Zuidema, Shell Oil Co. (Submitted by C. D. Johnson, Sec'y.)

WHEELING, November '55 meeting. R. McCray, The Garlock Packing Co., presented a paper entitled "Oil Seals." (Submitted by S. J. Litten, Sec'y-Treas.)

(Lube Abstracts, from p. 47)

physical behavior of lubricant mixtures. Possibility of simplifying phosphating practices for the cold drawing of wire, strip, etc., in the light of new developments. Problem of lubricants for cold working of stainless steels. (Order No. 3551, Price $4.80)

Friction, Wear & Lubrication in Wire Drawing, by W. Papsdorf; "Stahl Und Eisen," Vol. 72, No. 8, 1952, pp. 393-399; 10 figs., 2 tables, 4,700 words. Explanation of friction conditions, based on latest research results. Conditions of lubrication in wire drawing and Heidebroek's concept of boundary friction. Dependence of coefficient of friction on velocity of sliding and on surface pressure. Scoring of wire-drawing dies; die wear. Dependence of viscosity of fatty oils and mineral oils upon temperature and pressure. Lubricants used in wire drawing. Importance of lubricant carrier, wire coating and correct lime: soap ratio, particularly with metal soaps. (Order No. 3050, Price $9.40)

Diminution of Plastic-Deformation Zone in the Machining of Metals under the Action of Surface-Active Liquids, by G. I. Epifanov & L. A. Shreiner; "Doklady Akademii Nauk SSSR," Vol. 80, No. 5, 1951, pp. 781-782; 4 figs., 1,400 words. Experimental study of influence of a surface-active liquid (ethyl laurate) upon the size of the zone of plastic deformation produced in surface layer of an aluminum specimen during machining. Specimen preparation; method of developing structure of specimen by recrystallization and etching. Particulars on extent and nature of zone of plastic deformation obtained in dry machining as against machining with a surface-active liquid (Figs. 1 & 2). Depth of zone of plastic deformation as function of layer thickness removed in dry vs. active-liquid machining. Effect of surface-active cutting media on degree of work hardening in machining. (Order No. 2910, Price $2.80)

Aqueous Colloidal Clay Slurries as Lubricant-Coolants in the Deep Drawing of Metals, by S. Ya. Veiler, N. N.

Petrova & P. A. Rebinder; "Izvestiya Akademii Nauk SSSR," Otd. Tekh. Nauk, No. 4, 1946, pp. 625-628; 4 tables, 1,800 words. Avoidance of metal-to-metal contact in deep drawing operations. Results obtainable with a small addition of a highly colloidal solid filler to liquid drawing lubricants. Two groups of fillers: inert (roundish) and active (flaky shapes). Examples of hydrophilic and oleophilic active fillers. Author's laboratory and plant studies of bentonite or Florida clays in the form of water-diluted colloidal suspensions. Particulars on preparation of clay slurries; precautions to be observed; particle size ranges of clay fractions. Performance of clay slurries when added to conventional soda-soap solutions in industrial deep drawing of brass and bimetals. Numerical data on tool wear, number of defectives, and surface finish of product. Optimum quantity of clay slurry addition. (Order No. 2429, Price $3.60)

Chemical Coatings for Improved Cold Shaping of Steel, by W. Overath; "Stahl Und Eisen," Vol. 68, No. 13-14, 1948; 4,300 words. Technological importance of chemical coatings, especially metal phosphate coatings, for the cold shaping of steel. Phosphating practices. Explanation of phenomena observed in the most varied cold shaping operations, on the basis of the chemical and physical properties of metal phosphate coatings. Data on usefulness of phosphate coatings in the drawing of wire of high tensile strength, bar stock, etc. Advantages of phosphate coating: Reduction of work of deformation; increase in number of passes without process anneal; improved surface quality of product; longer life of drawing dies; less expensive lubricants. (Order No. 2179, Price $7.55)

Properties of Steel Wire Drawn at Increased Speed & Temperature, by F. Schwier; "Felten & Guilleaume Carlswerk Rundschau," No. 25, May 1939; 21 figs., 3,050 words. Results of experiments intended to determine the influence of increased drawing speed and increased drawing temperature on pull required, operating conditions, and mechanical properties of wire. Testing arrangement. Improved lubricating effect at higher temperatures; explanation of this result by the example of rape oil. Viscosity vs. lubricating power of lubricant. (Order No. 1021, Price $3.75)

New Process for Making High-Speed Steel Wire, by K. Przheblevski; "Kachestvennaya Stal," Vol. 5, No. 12, 1937, pp. 45-47; 6 tables, 2,250 words. Development of new technical process for rolling and drawing of high-speed steel wire. Cutting annealing time in half by isothermal heating. Elimination of cold work . . . lubricants for drawing; drawing speed; maximum drafts, etc. (Order No. 683, Price $2.25)

(New Products, from p. 36)

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Lubrication in the News

Abandonment of Master Saybolt Viscometers as Standards. The National Bureau of Standards announces its intention to abandon after June 30, 1956, the use of the master Saybolt viscometers as standards for the calibration of Saybolt Universal and Saybolt Furol viscometers. Saybolt viscometers are widely used in the petroleum industry to measure viscosities of lubricating and fuel oils.

Many years ago, these master instruments were adopted in cooperation with the American Petroleum Institute's Committee on Viscosity Standards, as representing Saybolt instruments having normal flow times. The master instruments have been in the custody of the National Bureau of Standards, where they have been used to determine corrections to be applied to the flow times of instruments submitted for certification.

More recently, the American Society for Testing Materials has published standard conversion tables and factors for the conversion of kinematic viscosities to Saybolt viscosities. Because of the greater precision possible in determining kinematic viscosities, these tables and factors provide more precise definitions of the Saybolt viscosity scales than do the master instruments. Consequently, continued use of the master instruments has become unnecessary and technically undesirable. It is therefore proposed to abandon them as standards after June 30, 1956.

Any objections to this proposed action should be brought promptly to the attention of the Director of the National Bureau of Standards.

Current Literature

"Determination of Shear Stability of Non-Newtonian Liquids," Special Technical Publication No. 182, reports the data from a cooperative effort to develop a suitable test method for measuring shear stability of non-Newtonian liquids. Test methods studied include sonic vibrators, sharp-edged orifices, flow through a diesel fuel injector, and a pressure release valve. (American Society for Testing Materials, LE-12/1, 1916 Race St., Phila. 3, Pa., 75¢ per copy.)

"Using Fatty Acids in Lubricating Greases." Factors improving the uniformity and performance of lubricating greases, as influenced by the fatty acid or soap employed, are described in this new bulletin. The advantages of fatty acids, as a replacement for tallow and other vegetable fats (glycerides), are summarized and various examples of improved grease formulations suggested. (AASGP, Fatty Acid Div., LE-12/1, 295 Madison Ave., New York 17, N. Y.)

"1955 Collected Technical Papers, ASTE" (347 pages), presents in one volume all of the papers and panel discussions presented at the 23rd annual convention of the American Society of Tool Engineers. Covering 16 separate lectures, four panels, and an ASTE Research Fund report on workpiece temperature distribution, a variety of essential subjects are discussed including: the use of magnesium for low-cost tooling, coding and classification systems, cold steel extrusion tooling, ce-

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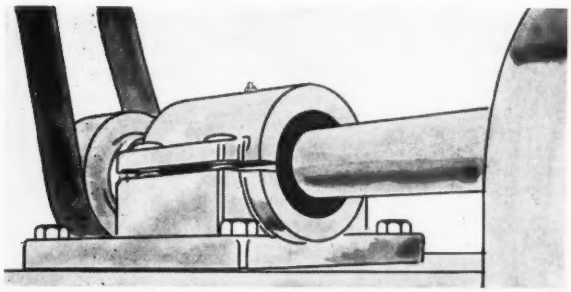

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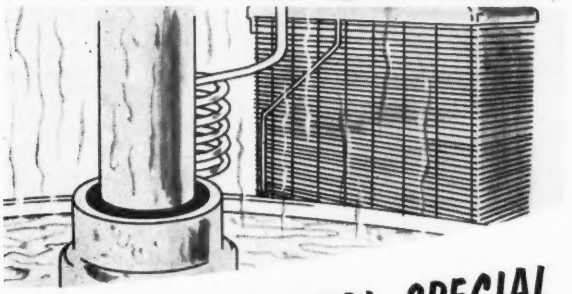

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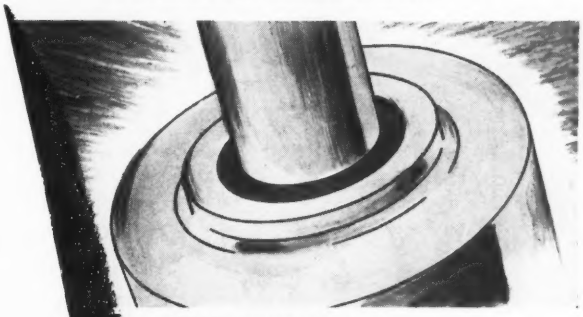

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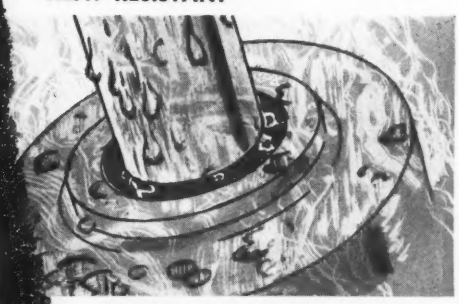

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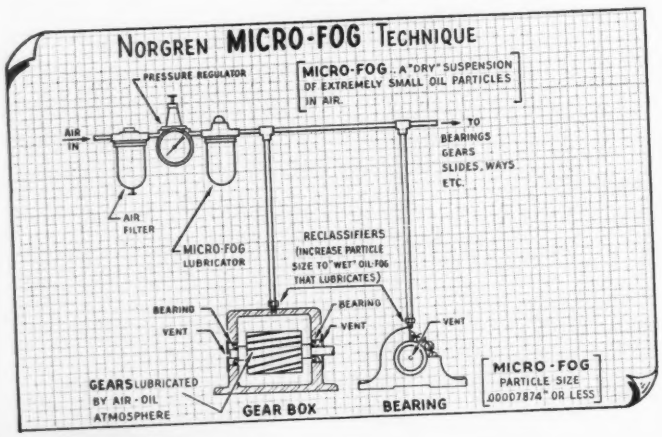

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Kinematic Viscosity Bath. Bulletin 743 describes the redesigned Model "S" kinematic viscosity bath. In addition to the applications of the bath, the bulletin discusses such new features as improved temperature uniformity, higher control sensitivity, interior visibility, and construction. (Precision Scientific Co., LE-12/1, 3737 W. Cortland St., Chicago 47, Ill.)

"A List of 'dag' Dispersions for Industry" (Revised), lists 42 colloidal and semi-colloidal dispersions for operational functions, maintenance, lubrication, machine design, and other industrial applications. Products include dispersions of graphite, molybdenum disulfide, mica, vermiculite, zinc oxide, and acetylene black; carriers and diluents are given for each product, along with typical applications and important physical data. New on the list is 'dag' Dispersion No. 224, a mica-and-water product for use as a metalworking lubricant (particularly for extruding aluminum) and as a dielectric coating. (Acheson Colloids Co., LE-12/1, 1630 Washington Ave., Port Huron, Mich.)

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quirements that have resulted from changes both in automotive engineering design and in petroleum products used in automotive service. (The Chek-Chart Corp., LE-12/1, 33 E. Congress Pkwy., Chicago 5, Ill., $2.50 per copy.)

Quick Connect/Disconnect "H" Coupling. Bulletin 240 describes a new quick connect/disconnect "H" Coupling, with particular reference to an exclusive U-packer which gives positive seal with compression set because of rubber distortion. Complete dimensions of all "H" Coupling parts, as well as ordering information for spare parts and complete couplings, are listed. (Snap-Tite, Inc., LE-12/1, Union City, Pa.)

"Physical Properties of Synthetic Organic Chemicals" (1956 edition), presents the latest data on more than 350 organic chemicals, and features 32 new products available from Carbide & Carbon. Arranged by related groups, with condensed data on applications, physical properties are given in tabular form, and an alphabetical index is included for easy reference. (Booklet No. F-6136, Carbide & Carbon Chemicals Co., LE-12/1, 30 E. 42nd St., New York 17, N. Y.)

"Molykote Types 165X & LOEX." Bulletin 105 (revised) describes in technical detail Molykote Type 165X, the open-gear lubricant; and Molykote LOEX, a lithium-base lubricant with a temperature range from —100 to 250 F. Also included is test data which compares "165X" with a widely-used, conventional open-gear lubricant in terms of frictional torque versus jaw load on a Falex testing machine. (The Alpha Molykote Corp., LE-12/1, 65 Harvard Ave., Stamford, Conn.)

"1956 Reference Guide to Dow Corning Silicone Products," describes almost 150 of the most generally used silicone products, 18 of which were first introduced within the last 12 months. As in the past, the products are grouped by physical form (fluids, compounds, greases, resins, and rubbers) and cross-indexed by usage (dielectrics, defoamers, damping media, etc.), enabling them to be located by what they do, as well as by what they are. Product descriptions are condensed, fact-packed, and devoted to essential data. (Dow Corning Corp., LE-12/1, Midland, Mich.)

"Stearates, Their Properties & Uses," lists the composition of 46 metallic stearates, shows graphically the industries in which they are used, and presents an analysis of recommended applications. Other helpful data include an explanation of Witco's analytical procedures, government specifications for metallic stearates, shipping information, and a guide to other Witco products. (Bulletin No. 55-2, Witco Chemical Co., LE-12/1, 122 E. 42nd St., New York 17, N. Y.)

"Filtration Manual For Product Designers." A comprehensive filtration manual is available describing applications and design considerations such as flow rates, viscosities of fluids, and contaminant removal, as well as filter costs, design space requirements, filter element types, and the selection of filter elements. A glossary of filtration terminology is included together with numerous illustrations and charts. (Dept. FM, Purlator Products, Inc., Rahway, N. J. Enclose .25c for postage and handling.)

Patent Abstracts

(Compiled by Ann Burchick, Aluminum Co. of America.)

Extractive Crystallization Processes by Means of Urea, Patent No. 2,710,855 (Alfred Champagnat, assignor to The British Petroleum Co., Ltd.) A process for the extractive crystallization of mineral oils which comprises treating a mineral oil with a solution of urea in a solvent comprising water, in the presence of biuret in the proportion of 0.2 to 5% by weight relative to the urea content, whereby a solid urea adduct is formed with some components of said mineral oil, separating said solid urea adduct from the remaining components of said mineral oil, decompos-

BENTONE* 34
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BATTENFELD GREASE & OIL CORP. OF CALIFORNIA, Compton, Calif.
BATTENFELD GREASE & OIL CORP. OF NEW YORK, North Tonawanda, N. Y.
BEL-RAY CO., INC., Madison, N. J.
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JAMES W. DOYLE CORP., Detroit, Mich.
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KEYSTONE LUBRICATING CO., Philadelphia, Pa.
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OIL DISTRIBUTORS OF PHILADELPHIA, Philadelphia, Pa.
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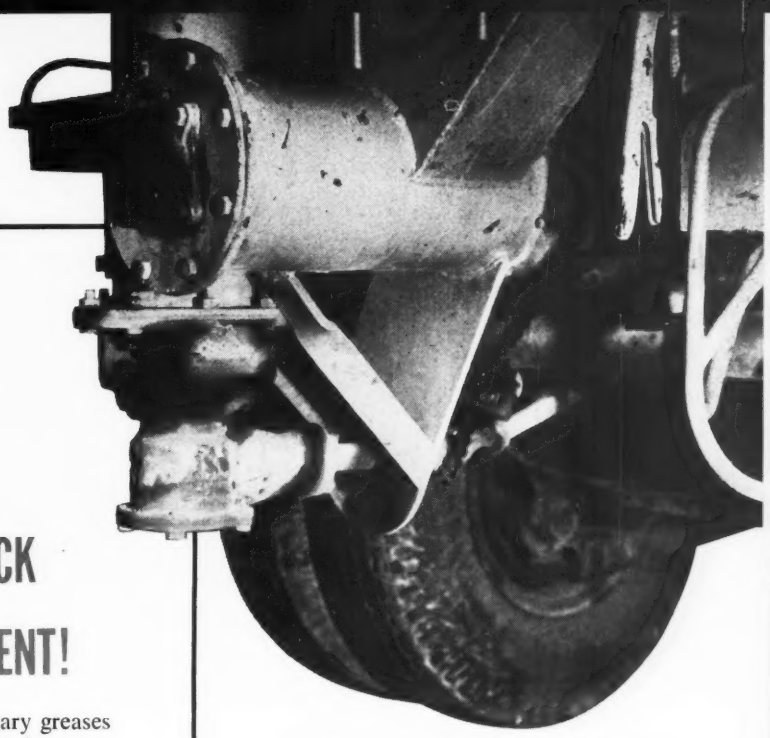
Exposed bearings are protected from any weather damage by Bentone grease



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FLEET OPERATOR SAVES $204 PER TRUCK IN BEARING REPLACEMENT!

Service Lumber Company used ordinary greases in its bulk-handling truck fleet and had to replace four $8.50 conveyor system bearings in each truck every month. Then the firm changed to Bentone*-based grease. Result: *No bearing failure for six months* . . . a $204 savings per truck.

The operator reports Bentone*-based grease:

- **cuts grease consumption in half**
- **stays in place in all weather**
- **resists wash-out**

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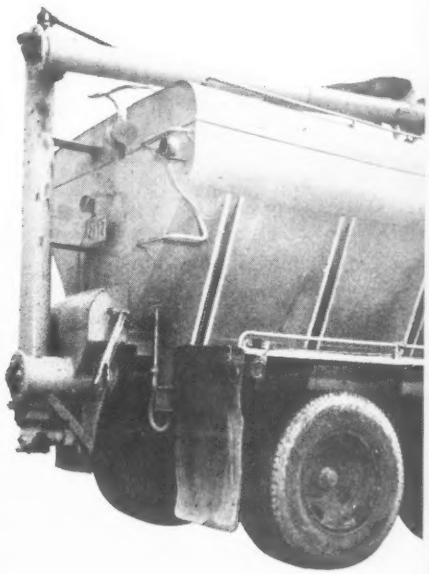
Truck operators report Bentone grease "stays put," resists extreme temperatures, dust, wash-out and abrasives



Texas fleet truck operator says Bentone grease cuts lubrication costs in half.

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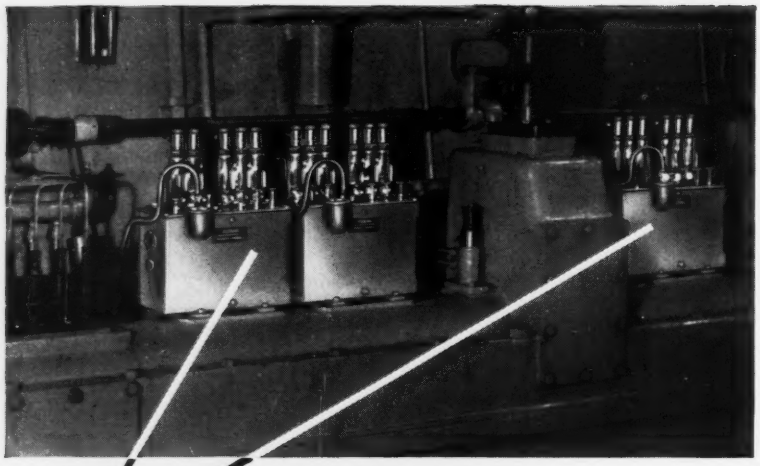

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- Gears of harder alloy for longer wear
- Every unit factory-tested before shipment

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ing said urea adduct and recovering the liberated components of said mineral oil.

Thickened Lubricant, Patent No. 2,711,393 (E. C. Hughes & E. C. Milberger, assignors to The Standard Oil Co.) A water-resistant thickened lubricant of good temperature susceptibility properties consisting essentially of a mineral lubricating oil of lubricating viscosity, an amount of an inorganic gelling agent to impart a grease-like consistency to the oil upon addition thereto, and an oil-soluble, water-insoluble, surface active alkyl alkylol imidazoline.

Inorganic Gel-Thickened Graphite Forge Die Lubricant, Patent No. 2,711,394 (F. Veatch & E. C. Milberger, assignors to The Standard Oil Co.) A stable uniform fluid lubricant composition containing as the principal and essential ingredients, a major proportion of a mineral lubricating oil, about 2.5 to about 35% powdered graphite, about 0.5 to about 5% of a finely-divided inorganic oil thickener, and having a viscosity of 300 to 1000 modified S.S.F. at 100 F.

Cutting Oil Composition, Patent No. 2,711,395 (A. Dorinson, assignor to Sinclair Refining Co.) Cutting oil compositions of improved corrosion and turbidity inhibiting properties which consist essentially of mineral oil and a sulfur-chlorinated pinene additive to which has been added basic barium mahogany petroleum sulfonate in an amount sufficient to inhibit corrosion and turbidity.

Cutting Oil Composition, Patent No. 2,711,396 (A. Dorinson, assignor to Sinclair Refining Co.) Cutting oil compositions, consisting essentially of a sulfur-chlorinated mineral oil base stock, to which has been added basic barium mahogany petroleum sulfonate in an amount of at least about 0.3 weight per cent based on the composition to inhibit corrosiveness.

Stabilized Hydrocarbon Fuel Oils Containing a Condensation Product of Ammonium Sulfide, Formaldehyde, and a Mono-Alkyl Phenol, Patent No. 2,711,947 (H. G. Smith & T. L. Cantrell, assignors to Gulf Oil Corp.) A fuel oil composition comprising a major amount of a hydrocarbon fuel oil and a minor amount, sufficient to stabilize the fuel oil against the formation of sludge and undesirable color bodies, of the product obtained by the process which comprises condensing in an aqueous medium one mol of ammonium sulfide and from 2 to 5 mols of formaldehyde at a temperature not exceeding 200 F. with from 2 to 4 mols of a monoalkyl phenol having from 4 to 12 carbon atoms in the alkyl substituent.

Lubricating Apparatus, Patent No. 2,712,364 (A. T. Laspe, assignor to

(Continued on p. 70)

E. S. P. S., Inc.

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Plant Engr. (399-LE) 50, CE; 4½ yrs. resp. for all plant maint. supv. power house; 5½ yrs. asst. to plant mgr. on plant budget; 3 yrs. on sales of production; 8 yrs. supv. office & planning staff on constr. projects; Midwest; $600.

Plant Engr. (400-LE) 45, MS, ME; 5 yrs. engr. charge of constr. of public, comm. & ind. bldgs.; 12 yrs. on design & assisting ch. engr. in food industries; U. S.; $7,800.

Ind. Mgmt. Engr. (401-LE) 34, ME; 2 yrs. supv. plt. engrg. dept. on measuring eqpt.; 1 yr. on plt. layout, time & motion studies; 1 yr. tool engr.; Midwest; $700.

Plt. Engr. (402-LE) 37, ME; 7 yrs. mech. engr. on steam generating plts. & resp. for plt. modifications; East or West; $6,500.

Ch. Engr. (403-LE) 54; 3 yrs. plt. engr. arsenal; 3 yrs. eqpt. engr. ordnance plt.; 1 yr. instructor in refr. & air-cond.; 8 yrs. refr. engr. in food plt.; U. S.; $7,200.

Plt. Mgr. (404-LE) 48, ME; 14 yrs. surveys, install. of eqpt., maint., & methods improvement in food processing ind.; 4 yrs. design & sold molded plastic chem. eqpt.; 8 yrs. process eqpt. tester & development in pkg. house; U. S.; $8,000.

Ch. Engr. (405-LE) 44, EE; 10 yrs. resp. for all utilities & plt. maint. (steam, elect., air, water & refrig.) for brewery & smelting plts.; 7 yrs. resp. for shift operation for utility; U. S.; $11,000.

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C-4302, Tech. Publications Unit Leader. 2 plus yrs. exp. in field servicing aircraft gas turbine powered plant; duties: preparation of tech. publications & reports relating to this work; strong possibilities for upgrading to position of greater resp. within next yr.; mfg. of engines; salary $7-8,000; loc. Mass.

C-4301, Sales Planner. 5 plus yrs. exp. in sales planning; duties: aiding developing marketing plan, integrating considerations of military commercial small aircraft gas turbine engines; mfr. of engines; salary $7-8,800; loc. Mass.

C-4300, Lubr. Engr. ME; age up to 35; 2 plus yrs. exp. in plant engrg. & preferably in some lubrication capacity; know: lubrication & petroleum products; duties: assisting in plant engrg. work at first & later work into lubr. work; mfr. brass; salary $5,400-7,200; loc. Chicago; empl. will pay fee.

C-4297, Sales Engr. Eng. degree preferred; previous exp. in design, sales or service of turbo jets and/or turbo props.; know: contract terms & conditions; duties: liaison between engrg. dept. & customer requirements in field on turbo jets & turbo props.; mfr. of jets; salary $7-8,000; loc. Mass.

C-4259, Gen'l. Mgr. Cylinder Gas Sales. 5 plus yrs. exp. directing national sales of allied products; duties: directing national sales of industrial gases, welding & cutting eqpt., & safety devices; for mfr. of gases; salary $12,000; loc. Penn.

E.S.P.S. Celebrates 30 Years in San Francisco

The Engineering Societies Personnel Service San Francisco office opened in January of 1925. ESPS, as a cooperative enterprise, was established by four national engineering societies in New York to help engineers find new or better jobs, and to help employers locate qualified personnel needed in technical work. Later, extended to San Francisco it has operated from the start in the same offices, Suite 715, Mechanics Institute, 57 Post Str., with the same manager, Newton D. Cook, 219 Linda Ave., Piedmont, Calif.

In thirty years, thousands of engineers of every kind and at every level have sought employment assistance; thousands of western engineering jobs in every technical occupation have been made available. San Francisco ESPS has filled 8,000 western engineering positions, probably fifty-million dollars in terms of today's average annual salary. San Francisco ESPS has long been established in the Western engineering community. Engineers placed work in every part of the world and in every branch of engineering: 2,500 Civil Engineers in construction, government, consulting; 2,000 Mechanical Engineers in industry, design, manufacture; 2,000 Electrical Engineers in laboratories, production, sales; 1,500 Mining Engineers in production, extraction, fabrication, reduction.

ESPS was established in 1918 as a non-profit service operating in the east after World War I to help engineer veterans find jobs in civilian life. Originally sponsored by the American Society of Civil Engineers, American Institute of Mining & Metallurgical Engineers, American Society of Mechanical Engineers, American Institute of Electrical Engineers, it now includes ten national or regional cooperating societies and organizations with a membership approximating 200,000; and it provides a nation-wide employment service available to 550,000 U. S. engineers at professional, technical, and sub-professional levels through offices in New York, Chicago, Detroit, and San Francisco.

Did you know that ESPS, Inc. is owned and operated by the American Society of Mechanical Engineers, the American Society of Civil Engineers, the American Institute of Electrical Engineers, and the American Institute of Mining & Metallurgical Engineers?

Did you know ESPS is also sponsored by six other National and local well-known societies?

Did you know ESPS is set up on a non-profit, self-supporting basis?

Did you know ESPS has offices in Chicago, Detroit, New York, and San Francisco?

Did you know ESPS has been serving the engineering and scientific professions since 1918?

(Continued from p. 68)
Lincoln Engineering Co.) Lubricant spraying apparatus comprising an injector having an inlet adapted for connection to a lubricant delivery line, a measuring chamber, and an outlet from the measuring chamber, a piston slidable in the measuring chamber, said piston having a retracted position wherein it is located adjacent one end of the injector and being movable from said retracted position toward the other end of the injector to deliver a measured charge of lubricant through the outlet upon increase of pressure in the inlet, the injector being adapted to recharge with lubricant upon relief of pressure in the inlet and the piston thereupon returning to retracted position, an atomizer having a lubricant inlet and an air inlet, an air valve mounted on said one end of the injector having an inlet and an outlet, a lubricant line connecting the injector outlet and the atomizer lubricant inlet, an air line connecting the valve outlet and the atomizer air inlet, and a pin extending from the piston through said one end of the injector into the valve, said pin controlling the valve to effect opening thereof for delivery of air to the atomizer whenever the piston operates to deliver a measured charge of lubricant to the atomizer, and holding the valve closed when the piston is in retracted position.

Pour Point Reduction, Patent No. 2,712,518 (S. C. Vaughn, assignor to Tide Water Associated Oil Co.) A process for reducing the pour point of hydrocarbon oils which comprises dissolving glyceryl-12-hydroxy-stearate in a hydrocarbon oil, cooling the resulting solution sufficiently to precipitate said glyceryl-12-hydroxy-stearate, and separating oil of lowered pour point from the resulting precipitate; said cooling and separation of oil being conducted at a temperature above the pour point of the oil being treated.

Hydrocarbon Oil Additive, Patent No. 2,712,526 (J. P. McDermott, assignor to Esso Research & Engineering Co.) A mineral oil composition having incorporated therein in the range of about 0.02 to 50% by weight of the salt formed by reacting a thialdine-type compound.

Improved Lubricating Greases Containing Dihydroxy Stearic Acid Soap, Patent No. 2,712,527 (L. A. Mikeska & A. J. Morway, assignors to Esso Research & Engineering Co.) A lubricating grease comprising a major proportion of a lubricating oil and a minor grease making proportion of a grease thickener containing a metal soap of dihydroxy stearic acid and a metal salt of low molecular weight aliphatic carboxylic acid having not more than 5 carbon atoms, said soap and salt being present in the form of a soap-salt complex and in a mol ratio within the range of about 1 to 3 mols of said low molecular weight aliphatic carboxylic acid salt for each mol of said dihydroxy stearic acid soap.

(Lube Lines, from p. 13)
brication. This ties in directly with the life expectancy of the machine. Statistics, of course, require records; and records require periodic check-up as to the condition of the machine, its productive ability, the efficiency of its lubricating system, and the performance of its lubricants.

When one buys an automobile today, there is the urge to follow a regular period of check-up: 500,

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10.0, 5000, and 10,000 miles; then repeat. In addition, it is customary to have intermediate check-ups at 1500- to 2000-mile intervals when the motor oil is changed and the chassis is lubricated. The modern automobile will average around $3,000.00 in cost. Some units of industrial machinery, in contrast, will cost several hundred thousand dollars. Obviously, these productive machines require equally as careful check-up as the family automobile, if the initial cost is to be protected over the useful life period. Depreciation and obsolescence determine the length of this period. The latter, naturally, is a function of progress and improvement. Depreciation, however, can be more closely controlled by lubrication check-up which, in reality, is a form of insurance.

The cost of this insurance, in terms of the annual cost of lubricants, is extremely low as compared with the value of the investment. Million-dollar machinery investments are not unusual. Often, less than one percent of this investment is paid out annually for lubricants. Compare this with a five percent write-off annually based on a useful life of twenty years.

Simplified Lubrication Recommendations. There is another trend in industry just as interesting to management as the trend towards lubrication statistics; it involves the trend towards simplification of lubrication recommendations. This is a commendable procedure in view of the practice which used to prevail, of carrying an unnecessarily-large inventory of lubricants in stock, often simply to conform with the variety of recommendations issued by certain machinery builders. The petroleum industry has been especially active in consolidating their lubrication recommendations in order to reduce large plant inventories of this nature. As a result, there are available today simplified lubrication recommendations for machine tools, construction machinery, and certain of the other heavy industries.

The simplified lubrication recommendation is invaluable to plant personnel who are responsible for check-up:

(a) The possibility of misuse or applying an unsuitable grade of lubricant is reduced.

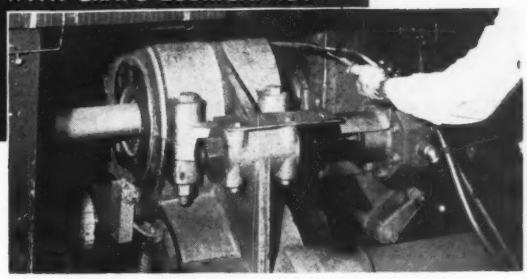
(b) Oils and greases of maximum stability are available to meet widely-varying conditions of operation.

(c) Storage and handling practices can be perfected so that there will be the least possibility of mix-up or contamination.

(d) Full advantage can be taken of the research endeavors of the petroleum industry to adapt selected combinations of petroleum base oils, synthetics, and soaps to modern precision machinery.

Housekeeping Practices. A well-organized schedule for lubrication check-up automatically develops good housekeeping practices around machinery. One of the outstanding benefits which result from the greater attendant cleanliness, is reduction of contamination of lubricants in storage, handling, and service. Clean oil-house and machine room conditions naturally reduce the possibility of

EXTRUSION MANUFACTURERS GAIN PRODUCT UNIFORMITY WITH GRAFO LUBRICANTS...



Here Grafo 120 is being sprayed on a die in a 500-ton Extrusion Press at a large aluminum extrusion plant.

Leading Aluminum Extrusion Manufacturers are finding Grafo Colloidal lubricants a great aid in gaining product uniformity. Grafo extrusion lubricants protect the dies in extrusion presses —prevent sticking and build-up in the dies—results, "quality products at lower cost." There's a Grafo Industrial Lubricant which will aid your operation. Write us today for information on your operations.

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fire and hazard to operating personnel. Plant insurance rates can be directly proportional to the extent to which these hazards exist. The Associated Factory Mutual Fire Insurance Companies of Boston have interesting records which indicate fire causes. In cotton mills, for example, lack of lubrication has been found to be one of the principal reasons. In other words, inadequate lubrication can mean overheated bearings, hot spots due to metallic contact, and sufficient vaporization of leaked oil or grease to cause a flash in case of spark or arcing.

In effect, the success of the check-up procedure can be measured by management in terms of its fire insurance rate. A poor rating indicates that the fire insurance company regards the plant as a poor risk. Lubricant leakage due to poor housekeeping is one of the most obvious fire hazards which can contribute to a poor rating.

(Men of Lubrication, from p. 15)

(Newton's) is all that is necessary. In those cases where it would be desirable to use Stokes' more accurate formula, it is likely that turbulence would exist and it would be inapplicable. It is interesting that the same problem of two concentric cylinders has led, in the hands of G. I. Taylor, to a valuable deduction on the onset of turbulence in this space ("Stability of a Viscous Liquid Contained Between Two Rotating Cylinders," by G. I. Taylor, Trans., Roy. Soc. of London, Ser. A, 1923). Petrov's formula was derived from Margule's equation for the concentric cylinder rotation viscometer.

Petrov was the author of a large number of technical papers; and for his outstanding work, particularly in railroad economics, he received much recognition in his own country. He was, during his life time, one of the most distinguished of Russian engineers.

ACKNOWLEDGEMENTS. The main sources consulted in the preparation of this article were: "Vojennaya Enciklopedija," Tome 18, p. 394; "Biographical Notes on Petroff," from Ostwald's "Klassiker," No. 218, 1927; and notes and private communications from Mayo D. Hersey.

(This is the seventh in a series of articles by F. R. Archibald, Analyst, of Arthur D. Little, Inc.)

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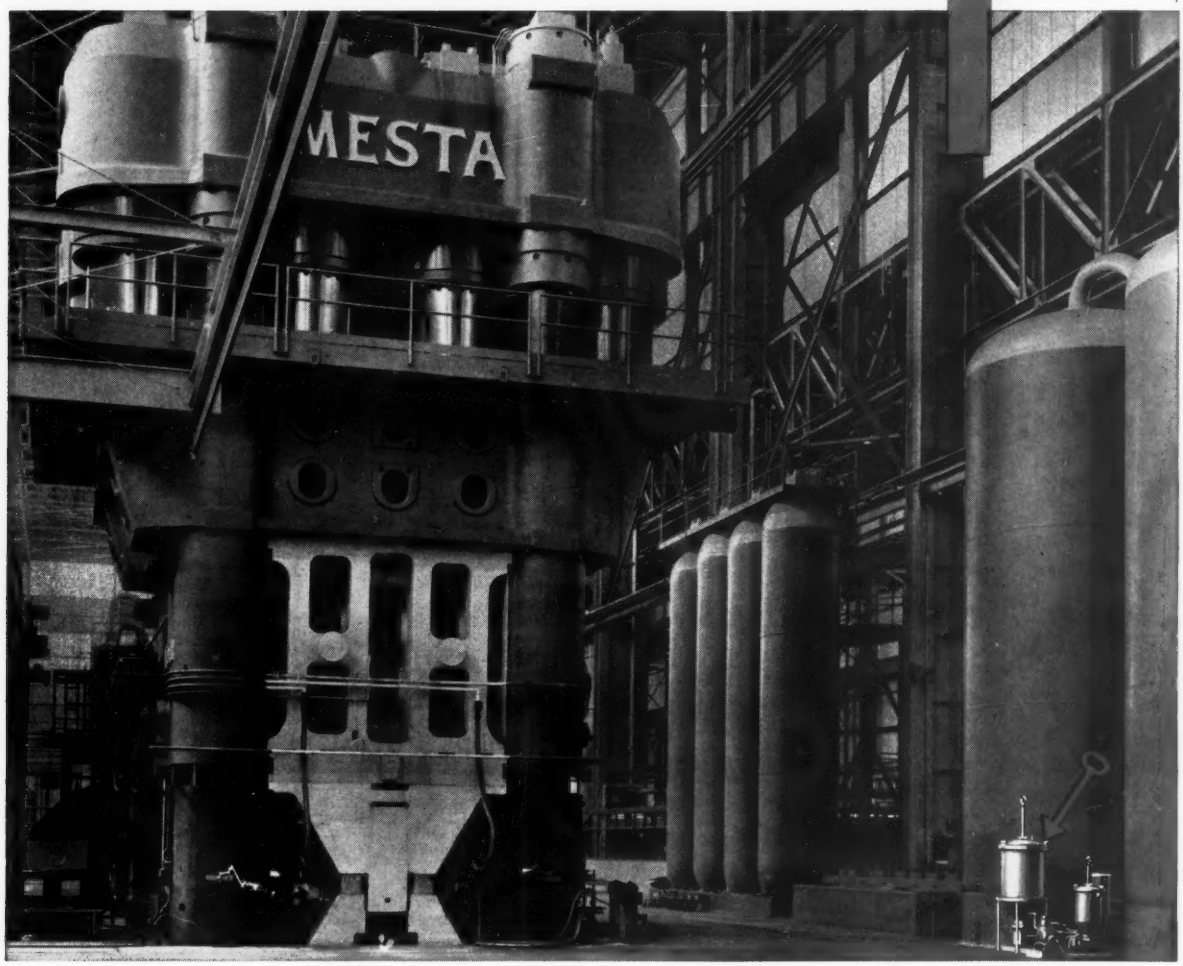

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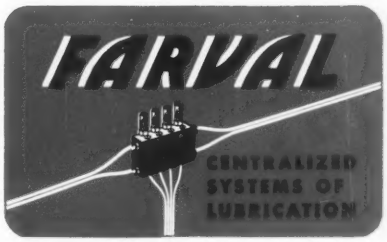

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